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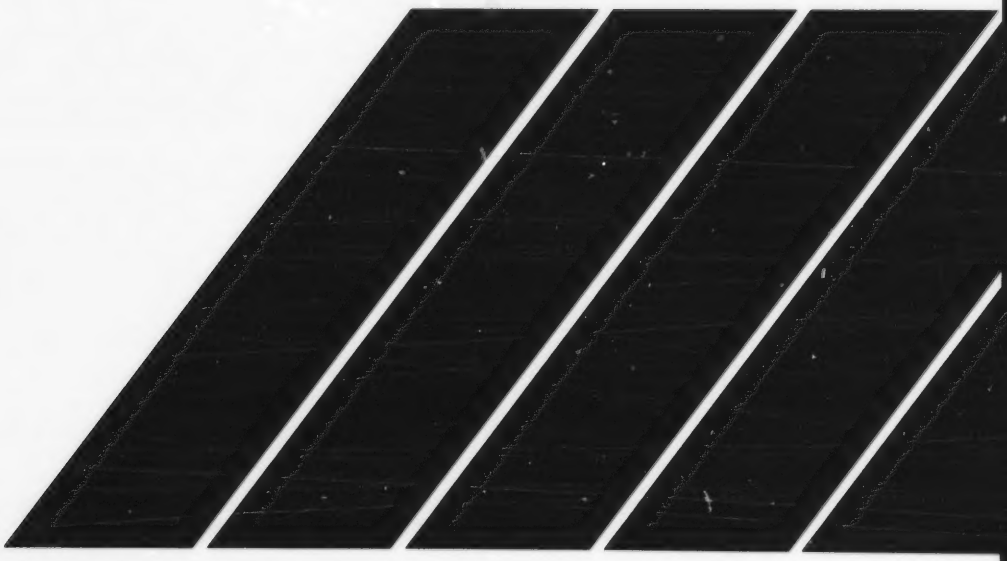

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# DEVELOPMENT OF A SECOND GENERATION LARGE VERTICAL AXIS WIND TURBINE GENERATOR: PHASE I

**PREPARED FOR:**

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Energy, Mines and Resources Canada
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**ABSTRACT**

This is the final report of a study intended to identify and appraise a second generation of large vertical axis wind turbines. The study has included three intermediate reports which have presented results of particular concern: Milestone One gave the results of a survey of wind resources, manufacturers and potential markets; Milestone Two described the two rotor and drive train configurations selected for further study; and Milestone Three presented a preliminary structural analysis and design of those two configurations.

The present report summarizes the three intermediate reports and presents appropriate cost estimates. These estimates point to total installed costs of a production 1-MW machine to have a lower bound of approximately $1,200,000. This cost could, however, be achieved only with a machine that incorporates features such as tailored airfoils, lightweight fibre composite blades and dynamically soft guy cables. Some of these concepts should be developed and proven on a smaller scale.

**RÉSUMÉ**

Le présent rapporte les résultats définitifs d'une étude qui avait pour but de déterminer et d'évaluer une deuxième génération de grandes turbines éoliennes à axe vertical. L'étude comprenait trois rapports intermédiaires faisant état d'aspects d'intérêt particulier. Le rapport de la première étape présentait un sondage des ressources dans le domaine éolien, énumérait les fabricants et décrivait le marché possible; le rapport de la deuxième étape décrivait la configuration de deux rotors et trains d'entraînement désignés pour de plus amples travaux de recherche; le troisième rapport consistait en une analyse préliminaire de la structure et de la conception des deux configurations.

Le présent rapport résume les trois rapports intermédiaires et présente une évaluation appropriée des coûts. L'évaluation comporte le coût total une fois installé d'un appareil pouvant produire 1 MW à un coût moindre d'environ 1 200 000,$. Ce prix ne serait cependent réaliste que si l'appareil était composé d'éléments tels des ailes fait sur mesure, des pales légères fabriquées de fibre composés et des haubans dynamiquement souples. Certains de ces concepts devraien faire l'objet d'éxperimentation à plus petite échelle.

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# 1.0 INTRODUCTION

## 1.1 Background

The past decade has witnessed a resurgence of interest in the harnessing of wind energy; it is seen as a plentiful source of clean and environmentally acceptable energy. The ratings of wind turbines have ranged from a few watts to about 4MW with the majority of utility-connected machines lying between 50 kW and 300 kW.

Claims have been made for greater cost effectiveness with greater size but, to date, all machines of ratings greater than 1MW have been prototypes and have not yet demonstrated this increased competitiveness. The 4MW Darrieus rotor wind turbine, "Eole", installed in 1987 in the Gaspé region of Québec is one such machine. It has demonstrated significant engineering advances but does not promise to be competitive with smaller wind turbines on a limited production scale. It has, therefore, prompted the question of what is the best configuration for a second generation of large Darrieus rotor wind turbines.

Attention has been directed at a rotor size of one half of the swept area of the Eole rotor with an electrical power rating of about 1MW. Such a machine could find broad acceptance by utilities and other large energy purchasers.

Lavalin Inc. is the owner and operator as well as designer of the 4MW Eole and is keen to utilize the understanding and expertise that it has gained. Future progress in the wind energy industry must be made with some care, with consideration of technical feasibility, reliability, cost, the market value of power, and the wind resource available. The overall purpose of this study is to prepare a basis on which future planning may be made. It is envisaged that subsequent phases of this program will include detailed design, fabrication, installation and testing.

The results of the present study, however, may suggest that the program be altered in some ways.

## 1.2 Objectives

The particular objectives of this study are:

1) to survey the world wide wind energy resources, potential users, manufacturers, and probable market value;

2) to analyse operational problems experienced by present wind turbines;

3) review the basic aerodynamics, structural criteria, and possible configurations;

4) to identify one or more configurations having a rating of about 1MW that should be examined;

5) to carry out preliminary analyses and designs for the selected configuration(s) to determine their structural viability;

6) to estimate the cost of producing one or twenty such machines;

7) to draw conclusions concerning the cost effectiveness of the proposed design(s), the level of reliability and to advise on the future steps in the program.

## 2.0 MARKET SURVEY

A considerable amount of information was gathered in the course of this project and was included as three volumes of appendices to the Milestone One report. The three volumes are:

Volume A1 - Wind Resources
Volume A2 - List of Manufacturers and Models
Volume A3 - Industry and Market Review

For full information these appendices must be consulted and only a brief summary will be given here.

### 2.1 Wind Resources

The wind resource maps presented in Appendix A1 have been gathered from a variety of sources and the terminology and format differ widely. Some measure wind energy by the mean annual wind speed and some measure it by the annual average kinetic energy flow (such as Watts per square metre). Neither measure is, by itself, entirely adequate for appraising wind energy potential for which the distribution of wind speed or energy flow is also needed.

The shape of the wind speed distribution is related to the type or cause of the wind. Trade winds are usually described by a Weibull distribution with a shape factor greater than 2.0 and most energy is associated with moderate wind speeds. Storm winds, often found at temperate latitudes, have a Weibull shape factor of 1.5-1.8 and more of the energy is at high wind speeds (15 m/s or greater). This information can influence the best speed at which the wind turbine should run for best energy yield and fatigue life.

The minimum annual wind speed required for financial viability of wind energy is approximately 6.5 m/s. The tables and maps in Appendix A1 show that there are many locations that meet this minimum wind speed but, as discussed later, this is not the only requirement for a viable wind energy project.

## 2.2 Manufacturers

Appendix A2 includes an alphabetical listing of 55 different manufacturers and their various models (of over 100 kW rating only). The distribution of these companies by country is given below.

| Country | No. of Manufacturers |
|---|---|
| Denmark | 14 |
| The Netherlands | 7 |
| United States | 6 |
| West Germany | 6 |
| United Kingdom | 4 |
| Canada | 3 |
| Spain | 3 |
| Japan | 2 |
| Sweden | 2 |
| Italy | 2 |
| Belgium | 1 |
| Switzerland | 1 |
| Greece | 1 |
| Austria | 1 |
| Hong Kong | 1 |
| India | 1 |
| | 55 |

The above list should not be interpreted as a level of activity in each country. For example, Belgium with one manufacturer has made and sold more wind turbines than Canada or Spain. In addition, some companies are inactive or have gone out of business since the list was made, and some companies are merely local assemblers of foreign machines. In the United States there is now only one true active manufacturer in this size range (greater than 100 kW).

Out of the 55 manufacturers listed there are 18 that have designs or prototypes of machines rated at 500 kW or more. However, there are probably only two cases of such machines being erected in numbers greater than three and those are in the 600-750 kW range.

## 2.2 Operational Problems

Appendix A3 contains the following sections.

A3.1 Review of Existing Products
A3.2 Operational Problems with Existing Machines

1.0 Introduction

2.0 Review of Operational Problems - Large Machines

3.0 Testing Conditions and Microsite Effects

4.0 California Commercial Wind Turbine Operating Experience

5.0 Conclusions and Recommendations

A reading of Appendix A3 confirms that the wind energy industry is not immune to the laws or nature and probability. What could go wrong has gone wrong, and projects based on too little design thoroughness and prototype testing have suffered accordingly.

The overall conclusions from this review are -

1) Problems can be classed into three divisions: early "infant mortality" failures, premature design and fatigue failures, and old age wearout. Prototype testing programmes may detect the first but not all of the second class.

2) Problems with larger more complex machines have, naturally, been more difficult and expensive to correct. There are few MW size machines with enough operating hours to confirm the specified design life.

3) Most production time has been lost due to structural and fatigue problems. Attention must be given to effects such as atmospheric turbulence. Prototype testing should be carried out at a high wind, turbulent location.

## 2.4 Markets

Appendix A3 of the Milestone One report contains the following sections.

A3.3 Potential Clients

1.0 Developing Countries

2.0 Developed Countries

2.1 Economics and Current Markets
2.2 Contacts in Developed Countries

The separation of countries into developed and developing is somewhat arbitrary but nevertheless useful in that it implies the presence or absence of an infrastructure and facilities.

### Emerging Nations

Quoting a study carried out in 1987 for the Solar Energy Research Institute, Golden, Colorado, 13 countries are identified as having the necessary conditions for a viable WECS installation. The requirements are both financial (suitable power rates) and technical (suitable wind regime).

In order to help the reader to understand the potential in each country a number of tables are presented in the appendix. These tables list for each country the present rate paid by consumers, the long term marginal costs of future conventional power and that of wind generated power.

The results of this survey should be interpreted with some care. The fact that Jordan is at the top of the list of 13 suitable countries does not mean that all of that country is a potential wind farm; nor does Argentina's absence from that list mean that there are not remote communities in that country with conditions that favour wind energy installations.

Industrialized Nations

Section 2.1 in Appendix A3 discusses the economics of a wind farm in California from which much can be applied to a similar operation elsewhere. The report points out that the cost of O&M (Operation and Maintenance) should not be underestimated and can be reduced to $0.01/kWh only for a very efficient operation.

In California the wind farm owner requires a purchase rate of at least $0.08/kWh. This is possible only through Standard Offer 4 contracts which are no longer available and which will expire in a number of years. The "avoided cost", which influences the purchase rate for new contracts is strongly tied to the cost of oil and is therefore very low at present ($0.027/kWh).

The only countries that have vigorous wind energy industries are those, such as Denmark, The Netherlands, and West Germany, which have programs to subsidize the cost of wind power

Utility Involvement

In some countries the electric utilities, whether publicly or privately owned, have taken a somewhat cool attitude to wind energy purchases. This has generally been the case throughout North American and Europe where utilities have regarded wind energy as unreliable and intermittent and have credited to it the value of the avoided fuel only. Only when the avoided cost of fuel is high, as it was in the early 1980's, has it been in the utilities' own interest to purchase power from wind turbines.

When it is to a utility's financial advantage, then it will consider not only purchasing wind generated power but also the ownership and operation of wind turbines. In West Germany this may occur within the next few years but elsewhere, such as in Denmark, utilities have become WECS owners due mainly to legislation, and public and political pressure.

It would appear unwise to develop a wind turbine solely for the "utility market" since a utility is likely to require greater proven reliability than would a private investor. In addition, even a utility will select a wind turbine, as with any power plant, by its cost effectiveness compared to other conventional and renewable power plants. Any viable wind turbine will, therefore, be compared with other wind turbines and their cost effectiveness.

Utilities are one group of possible clients for the manufacturer and are usually the final client for sale of power. Utilities must be educated about the advantages of wind energy (distributed, modular nature, low operating costs, etc.) and their cooperation must be sought.

## 2.5 Conclusions and Recommendations

This initial task of the project has brought together a great deal of information which is appended to the Milestone One report. The information cannot claim to be complete but it is sufficiently comprehensive to lead to the following conclusions.

1) Many prototypes and commercial models (mainly of HAWT's) have appeared over the last decade. Many have been discontinued for technical and/or financial problems.

2) The wind energy industry requires very thorough engineering and a methodical testing programme. The past is littered with failures.

3) The success of a wind energy project depends not only on the presence of an adequate wind regime but also on a market with assured purchase rates.

4) There has been a trend towards larger machines but, while there are prototypes to up to 3000 kW, commercial machines rarely exceed 300 kW.

5) Power purchase rates of at least $0.08/kWh are usually required for the economic viability of wind energy. Such rates are not normal in industrialized countries except where there is a programme to subsidize renewable energy.

6) There are many developing nations with the resources and the need for wind energy. There are also problems of installation facilities and maintenance.

7) Utilities may require greater reliability to become purchasers of WECS but they will still judge a machine on its cost effectiveness.

Wind energy and other renewable sources will understandably benefit from concern for the environment and recognition of the hidden costs of conventional energy. The timing of this recognition will vary from country to country and it does not appear that Canada is a present leader.

It cannot be said that there is a proven market for a 1 MW Darrieus rotor wind turbine. Nevertheless it is likely that there will be a market for almost any wind turbine that is shown to be reliable and which can be shown to be profitable for its owner. The competition for that market includes all conventional energy sources as well as HAWT's and other renewable sources.

It may be that Darrieus rotor turbines have an advantage in scale that HAWT's do not enjoy. The likely influence of scale should be considered in the subsequent parts of this study in conjunction with the parameters already established.

## 3.0 SELECTED CONFIGURATIONS

This section reviews the material of the Milestone Two report which covered design philosophy, design criteria, aerodynamics, and drive train design.

### 3.1 Design Philosophy

The overall objective is to determine how cost effective a large Darrieus rotor wind turbine can be made where the term "large" has been understood to mean a machine of approximately 1MW electrical rating. The goal must be to arrive at a machine that will equal or surpass the cost effectiveness of smaller machines (typically of 100-300 kW rating).

A common criterion used for HAWT's is that the total installed cost should be equal to or less than $1000 US per kW capacity. This is a very vague criterion since it does not mention the wind regime or the annual energy yield of the machine. A more meaningful goal is to measure the installed capital cost per annual kWh of energy produced. For example, for a windfarm in California to be financially viable the capital cost per annual kWh produced should be not more than $0.40 US.

It can be shown that when a rotor of a certain design is scaled up and the same design criteria must be met (such as limiting fatigue stresses or avoiding buckling), the mass of the rotor will increase as the cube of the size, whereas the energy captured increases only as the square of the size. Therefore, since mass is often a direct measure of cost when similar materials are used, it is difficult to make a large machine as cost effective as a smaller machine. The way in which this challenge can be met is by adopting different and more efficient configurations, materials, and construction techniques.

The approach adopted in this project was to present two designs. One uses a more conventional configuration and construction, while the second adopts designs and techniques that may not be proven but which do offer potential improvements.

### 3.2 Aerodynamics

The objective of any wind turbine is to capture as much energy as possible while minimizing the capital cost. It is, therefore, necessary to be able to predict the performance of the rotor and also, for subsequent structural analysis and design, to predict the distribution of forces along the blades.

The most common tool for carrying out these aerodynamic calculations is the double multiple stream tube (DMST) model, which is described in Reference (1). This model was included in the computer codes used in this project.

The performance of a rotor is strongly affected by the type of airfoil profile and its associated lift and drag characteristics. It is also affected by the control strategy used and, of course, by the swept area of the rotor.

Shapes, Sizes and Strategies

The combination of variables can lead to an overwhelming number of possible configurations and scenarios, and some assumptions were initially made for the "conventional" rotor (#1). The height-to-diameter ratio was selected as 1.50 which is the same as that used by the Eole rotor and some other Darrieus rotors. The same rotor solidity (NC/R where N=number of blades, C=blade cord and R=maximum radius) of 0.15 was also adopted.

The NACA0018 airfoil was chosen in preliminary investigations since this was also the airfoil shape used on Eole. In addition a constant chord dimension was assumed.

The "rated" power of the rotor can be defined as the maximum power produced but if no distinct maximum occurs in the power vs windspeed curve then the definition must be modified. In this study the rated power was defined as the maximum power occurring at a mid rotor windspeed of less than 18 m/s.

Three different swept areas were considered: $1500m^2$, $2000m^2$, and $2500m^2$. In addition, schemes involving a constant rotor speed or using a variable speed to limit the power output were considered.

The use of a variable speed can result in higher energy output but it has the following disadvantages:

1) a variable speed system, such as a frequency converter and control system, is required;

2) due to the inertia of the rotor, it is not possible for the speed of the rotor to track windspeed changes of less than about 15 seconds. It is, therefore, inevitable that the rotor will see frequent peak powers much greater than the rated power;

3) the rotor, the gearbox, the generator and the brakes must all be sized for the peak power output values;

4) the rotor must be free of serious resonant conditions throughout the range of operating rotor speed.

For these reasons the variable speed concept was not adopted. It should be noted that the concept of dual speeds was not rejected; this scheme requires a doubly wound generator and a second, lower rotor speed to be resonance free.

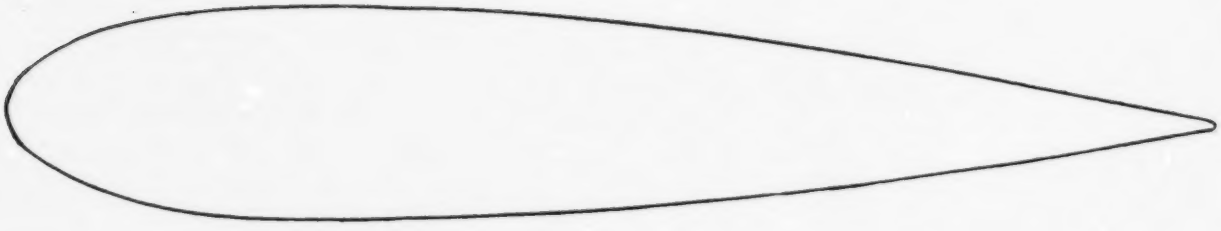
## Airfoils

The airfoils that have been used on most Darrieus rotors have belonged to the NACA00XX series and have had either 15% or 18% aspect ratios. These airfoils combine low drag with a high peak lift coefficient. The outline of a NACA0018 airfoil is shown in FIgure 3.1(a) and the lift coefficient vs angle of attack curve is included in Figure 3.2 (for Reynolds number of 4.E6).

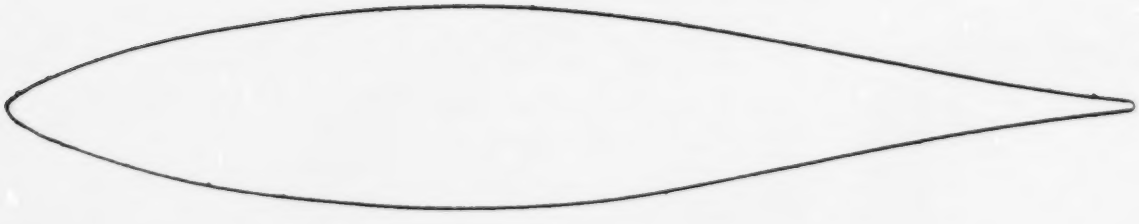
The NACA00XX airfoils were not designed especially for vertical axis wind turbines and their high peak lift coefficients result in high maximum output powers. Since the peak power is directly related to the rating and cost of most drive train components and not necessarily to the total energy captured, an airfoil shape with a lower maximum lift coefficient was recently designed. The design was funded through Sandia National Laboratories, New Mexico, and was carried out largely at Ohio State University. The profile of the 18% airfoil designed for laminar flow up to 50% chord is shown in Figure 3.1 (b).

The lift and drag coefficients of both the NACA0018 and the SNLA1850 at Re=4.E6 are compared in Figure 3.2. It is apparent that the peak lift coefficient has been reduced from 1.3 to 0.95 but that this has been at the expense of an increased drag beyond a 10 degree angle of attack. In addition, the predicted very low drag on the SNLA1850 is due to the maintenance of a laminar flow boundary layer over the leading portion. If surface roughness disturbs this situation then the airfoil may not enjoy this low drag. This leads to the possibility of combining the two types of airfoil in one rotor so as to use the advantages of both.

The laminar or "tailored" airfoil has been tested by Sandia National Laboratories on a 17m diameter rotor in Albuquerque and more recently on a 34m rotor in Bushland, Texas. Measured performance curves (2) largely validate the predicted behaviour of these airfoils.



NACA0018 airfoil



SNLA0018/50 airfoil

Figure 3.1. Comparison of Airfoil Shapes

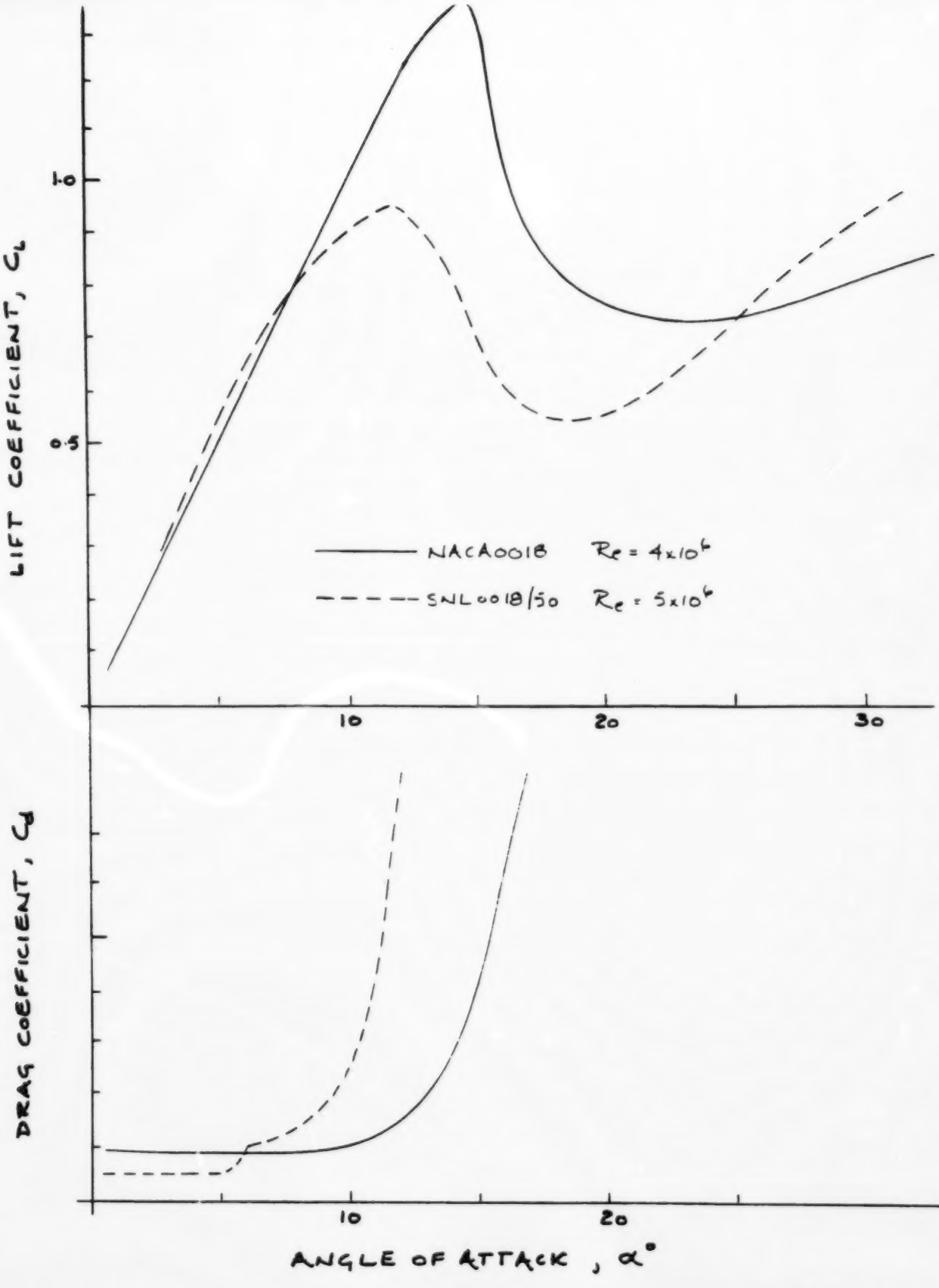



Figure 3.2. Comparison of Lift and Drag Coefficients for Two Airfoils

## 3.3 Rotor Configuration

Following the methodology explained in Section 1.3 two rotor configurations were selected. The first "conventional" rotor emerged from the assumptions listed in Section 3.1 with a swept area of 2000$m^2$ and an electrical rating of approximately 1M. The second "lightweight" rotor was designed to capture the same annual energy as the first and to include certain innovative features. The principal variables for each are listed in Table 3.1 and a sketch of each rotor is shown in Figure 3.3 (a,b). The performance curve for each configuration is shown in Figure 3.4.

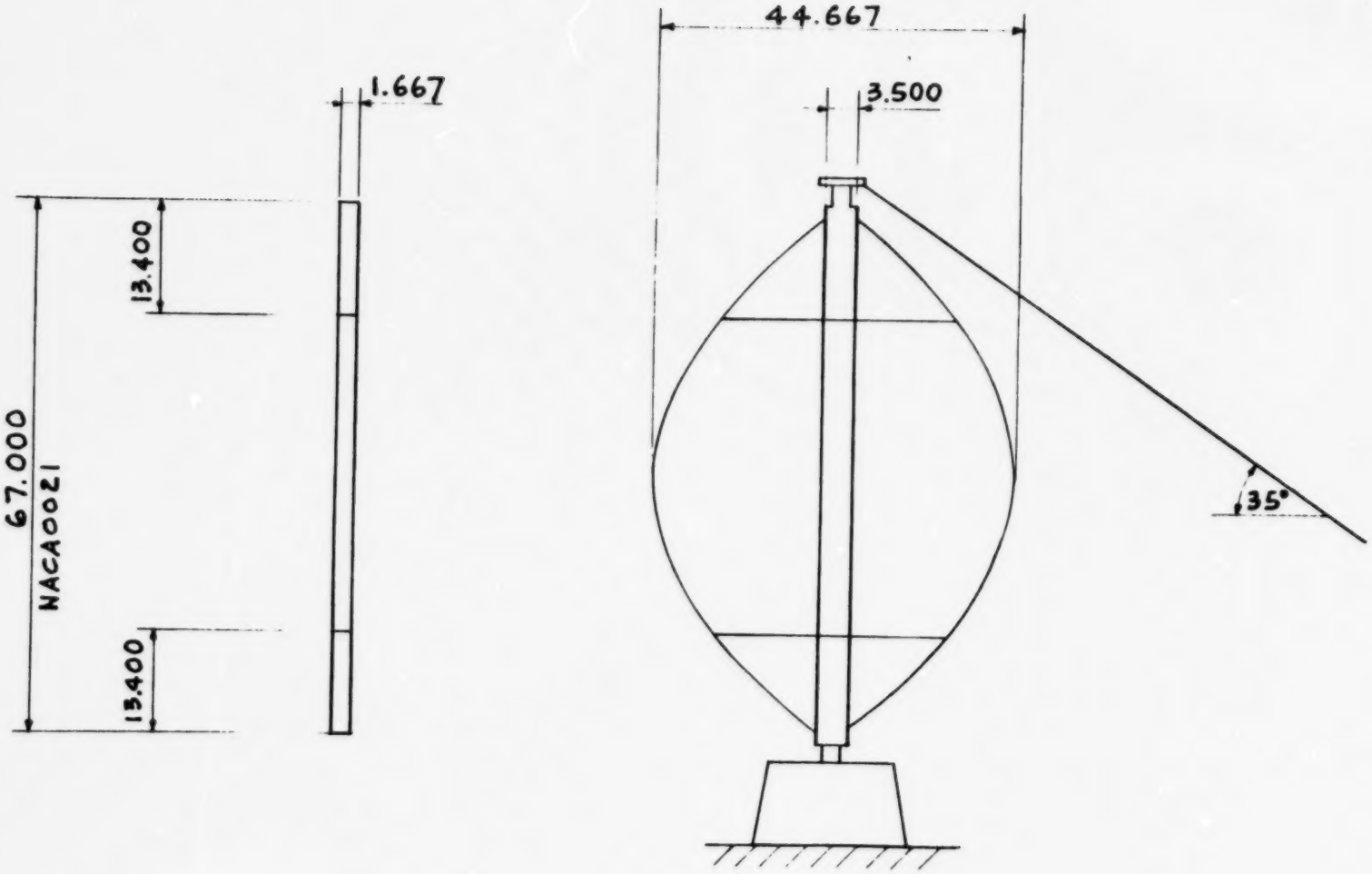



Figure 3.3(a) Selected Configuration. Rotor #1.

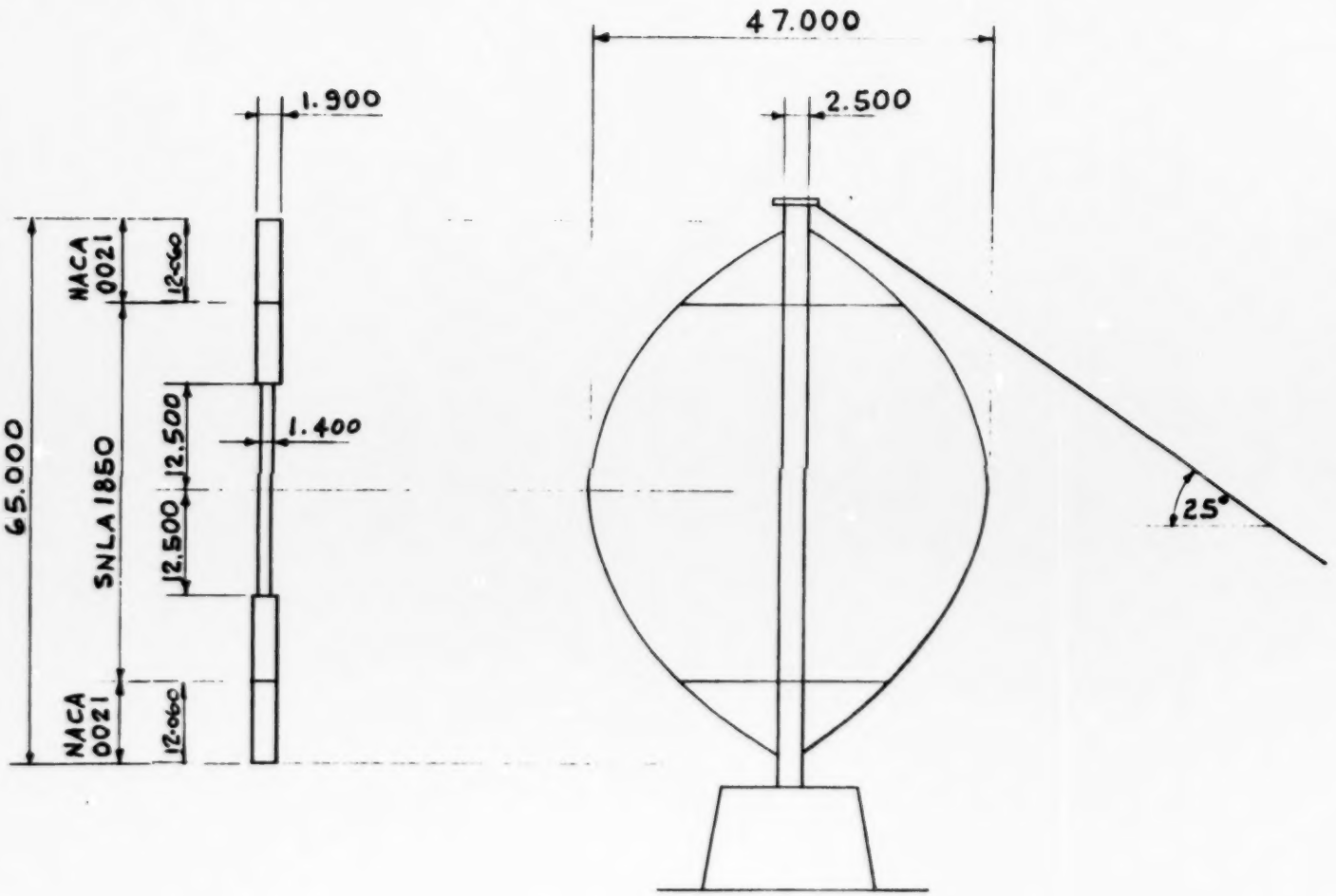



Figure 3.3(b) Selected Configuration. Rotor #2.

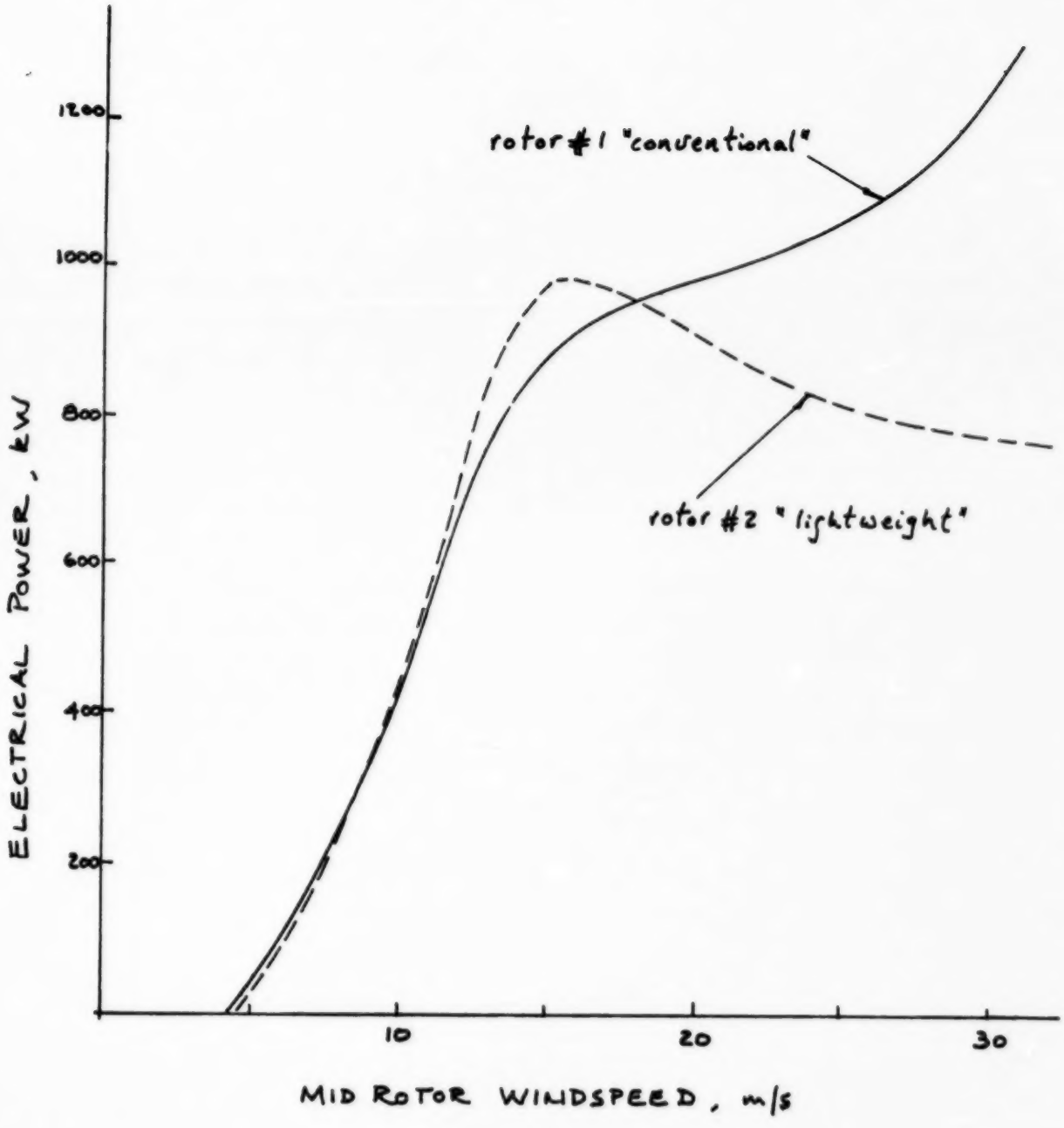



Figure 3.4 Performance Curves of Selected Rotors.

**TABLE 3.1**

**PRINCIPAL VARIABLES FOR TWO SELECTED ROTORS**

| | #1 Conventional Rotor | #2 Lightweight Rotor |
|---|---|---|
| .Height (between blade intersections) | 67.0 m | 65.0 m |
| .Diameter | 44.6 m | 46.4 m |
| .Aspect Ratio | 1.50 | 1.38 |
| .Swept Area | 1921 $m^2$ | 1975 $m^2$ |
| .Rotor Speed (constant) | 18.5 rpm | 23.5 rpm |
| .Airfoil(s) | NACA0021 | NACA0018/SNLA1850 |
| .Blade Chord(s) | 1.667 m | 1.40 m, 1.90 m |
| .Column Diameter | 3.50 m | 2.50 m |
| .Column Wall Thickness | 8.00 mm | 7.0 mm |
| .Location of Struts | 0.2, 0.8 height | 0.18, 0.82 height |
| .Guy Cable Size | 6 x 60 mm Ø | 6 x 20 mm Ø |
| .Guy Cable Pretension | 6 x 330 kN | 6 x 50 kN |
| .Rated Rotor Power ($V \leq 18$ m/s) | 1046 kW | 1072 kW |
| .Rated Electrical Power | 929 kW | 955 kW |
| .Rotor Annual Energy* | | |
| = 6.0 m/s | 1250 MWh | 1231 MWh |
| = 8.0 m/s | 2315 | 2396 |
| = 10.0 m/s | 3000 | 3157 |
| .Estimated Rotor Mass | 110 tonnes | 55 |

*Note:

1. A Rayleigh wind distribution is assumed, with a cut off at 18 m/s. All velocities refer to the midrotor. Sea level density (1.225 $kg/m^3$) is used.

2. Drive train energy loss is assumed to be 10 kW plus 10% of the aerodynamic power. Availability is assumed to be 100%.

3. Power and energy production were calculated using a double multiple streamtube model with Gormont dynamic stall (with a Masse modification). NRC dataset #13 (NACA0018) was used with Cd min =0.005 and SNL1985 data for the SNLA1850 airfoil.

## Horizontal Struts

The incorporation of horizontal struts between the column and the blade plays several roles. The struts transfer some, or most, of the torque into the column, they reduce the unsupported span of the blades and the accompanying bending moments, they stabilize the blade against buckling in survival winds, and they can be used to tune the natural frequencies of the rotor.

Horizontal struts do, however, have the following disadvantages: they add mass and cost, they add potential critical connections, they add aerodynamic drag and reduce the net energy output.

Their inclusion is a typical design problem that can be answered only by a series of careful studies. For some machines the question can be answered fairly quickly: is it possible, with the type of blade fabrication planned, to do without horizontal struts? For rotors of low solidity (relatively small blade chord) it is not structurally possible to do without struts. For these rotors it is then a matter of at what level the struts should be placed, their stiffness, and how should the connections be detailed.

For rotors of higher solidity or having blades with a tapered chord, it may be possible to do without horizontal struts. For such machines the upper and lower root connections usually become the critical location. In the present study both rotors are equipped with horizontal struts in a location similar to the struts on Eole (at 0.2 and 0.8 of the rotor height).

## Upper Restraint System

There are three possible systems of supporting the rotor and maintaining it erect.

1) Guy cables which are pretensioned and sized to give a sufficient restraint to the upper bearing.

2) Replacement of the guy cables by a rigid supporting structure.

3) Cantilevering the rotor from the base.

The last scheme has not been attempted on any Darrieus rotor with a rating greater than 100 kW. Even at that scale the system of bearings and inner stationary tube with an outer rotating tube was technically demanding. Larger machines using this scheme would suffer dimensional effects (bending moments would increase with the cube of the height) and this method was rejected.

The use of a rigid support to the upper bearing has the advantages that no prestressing is needed and that no vertical thrust need be transmitted into the rotor. The major disadvantage is that the members, if they are to act in compression, must be designed not to buckle. This implies a truss of considerable cross section and the total weight of steel makes the scheme uneconomic.

In summary, only the guy cable system is feasible on a large wind turbine and the direct and indirect cost of a cable system may be greatly reduced for the lightweight rotor where the pretensions are modest.

Blade Construction

A number of possible types of blades were considered. These were:

- formed sheet steel
- formed sheet steel with fairings
- steel plate core with fairings
- large aluminum extrusions
- light weight aluminum extrusions
- formed aluminum sheet
- aluminum plate core with fairings
- fibreglass pultrusions
- fibreglass windings
- fibreglass spar and fairings
- wood epoxy

All of these possibilities were judged by the following criteria:

- material cost
- assembly labour cost
- tooling cost
- ease of making connections
- ease of forming to troposkein profile
- fatigue strength
- weight

From the initial appraisal a short list of three possibilities was obtained. These were:

1) steel core with fibreglass fairings
2) large aluminum extrusions
3) fibreglass pultrusions

The steel core design was chosen for rotor #1 and a fibreglass pultrusion was chosen for rotor #2; sketches of both are shown in Section 4. The large aluminum extrusions were rejected on the grounds of material and tooling costs and difficulty of making fatigue resistant connections.

## 3.4 Drive Train

The earliest Darrieus rotors in the 1970's (of 5m to 25m diameter) all used anti-friction bearings, conventional gearboxes, and synchronous (induction) generators. Indal Technologies (DAF Indal) patented a drive incorporating a single large bullgear which also acted as brake disc and was supported on a hydraulic bearing. Adecon Energy Systems incorporated the lower bearing into the gearbox and placed the brake disc on the high speed end of the drive.

The 4000$m^2$ Eole machine uses a large diameter direct electrical drive (with no speed increaser) and a frequency converter system that allows the rotor speed to be varied as well as being able to act as an electrical brake. The subsequent German design of a similar machine uses a speed increaser but retains the variable speed feature.

### Alternative Configurations

It was mentioned in Section 3.1 that the rotor was likely to be operated at a constant speed. This implies that an electrical frequency conversion system is not essential although it still adds the ability to modify the rotor speed (useful for a prototype machine) and represents an alternative (and benign) braking system. The possibility of a direct electrical drive was eliminated due to the high cost.

The choices to be made in deciding the drive train configuration include the following:

- separate rotor pivot or incorporation into the lower bearings;
- same or separate support systems for bearing and gearbox;
- epicyclic or conventional gearbox;
- parallel or right angle drives for gearbox;
- incorporation of lower bearing into gearbox;
- the use of pneumatic or hydraulic brakes;
- the use of self adjusting brakes.

Over eight arrangements were initially considered but these were reduced to three which are illustrated (schematically) in Figures 3.5a, b, and c.

### Configuration (a)

The first arrangement, shown in Figure 3.1(a), features a combined bearing, gearbox, and generator assembly. The brake disc is located just above the bearing and is attached to the lower side of the turbo-flex coupling. This type of coupling has been used successfully on project Eole and is especially well suited to the high thrust load combined with limited pivoting motion.

In this design there is no need for couplings between the bearing and gearbox and between the gearbox and generator, but the main bearing must resist thrust, lateral and eccentric bending loads. In addition the components must be designed to fit together and any replacements must be equally compatible. Maintenance on one component may also affect adjacent components.

Configuration (b)

Figure 3.1(b) shows an arrangement in which all the components have separate supports and are connected through flexible couplings. Despite the need for these couplings the components can be selected and maintained independently and the main bearing does not have to resist the eccentric moment load.

The additional length of the low speed shaft introduces torsional softness into the drive train which can be important in the reduction of torque ripple during operational and in reducing torques due to accidental electric transients. The total enclosure space required for this configuration may be higher than for alternative arrangements.

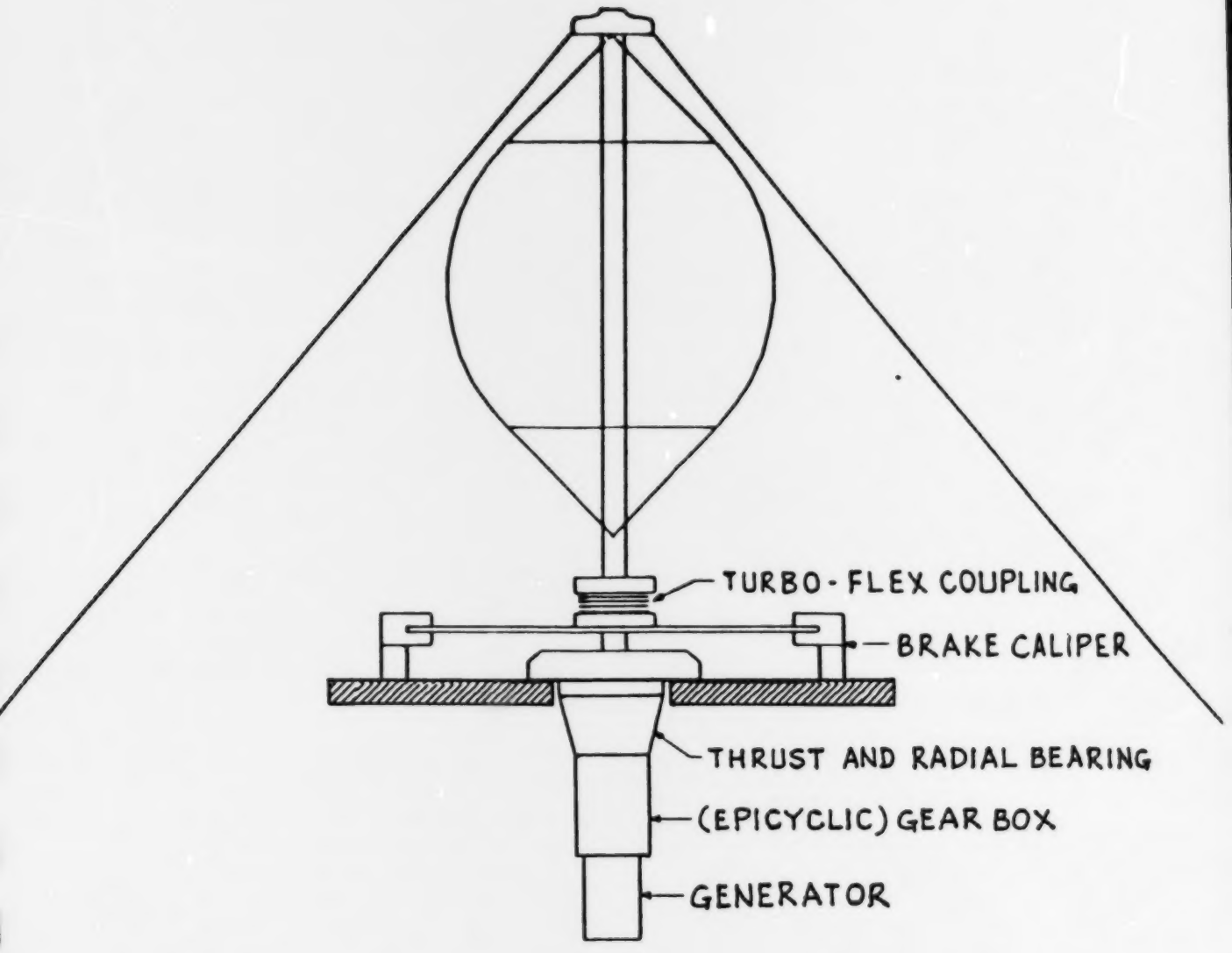



Figure 3.5(a) Alternative Drive Train Configurations.

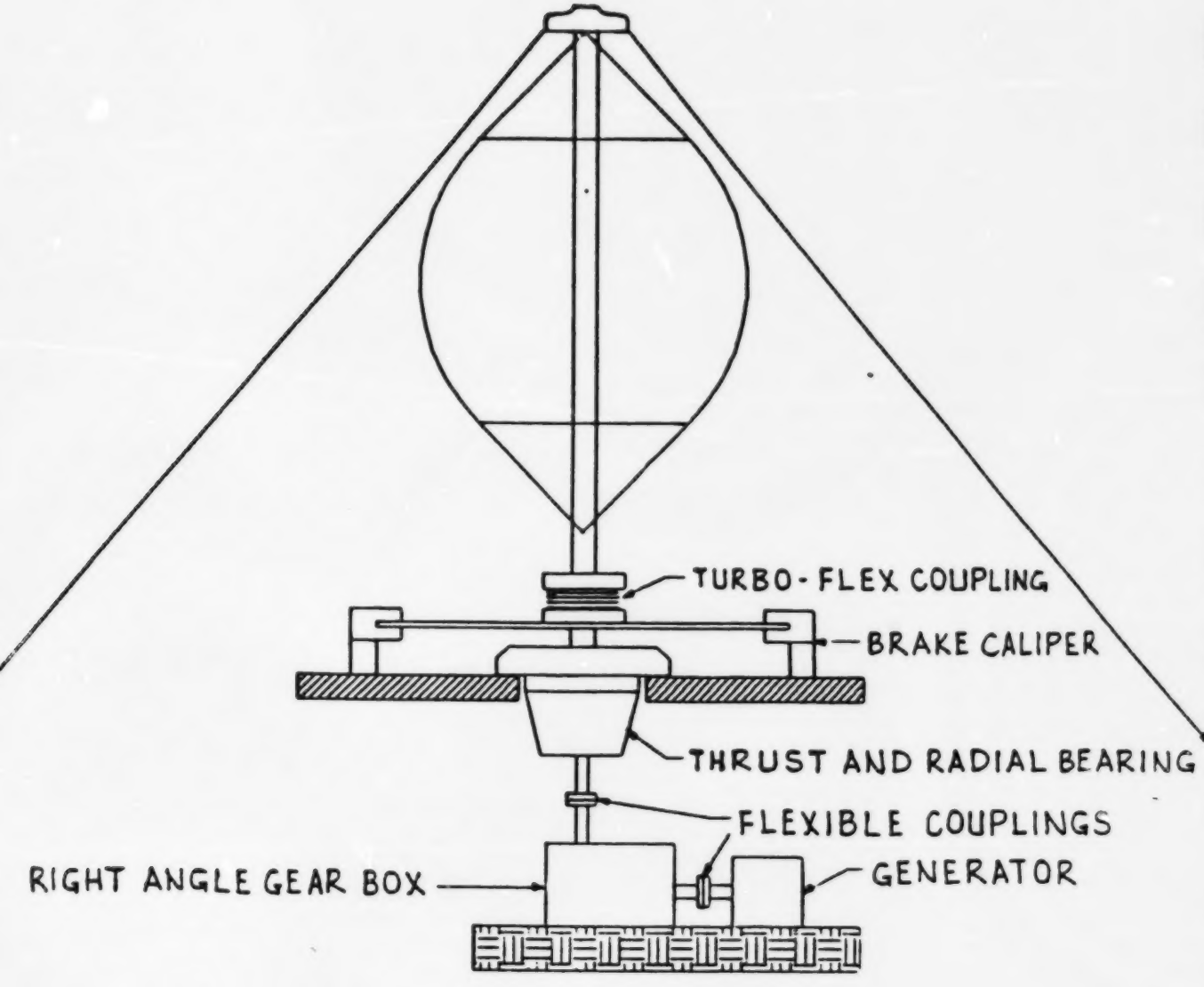



Figure 3.5(b) Alternative Drive Train Configurations.

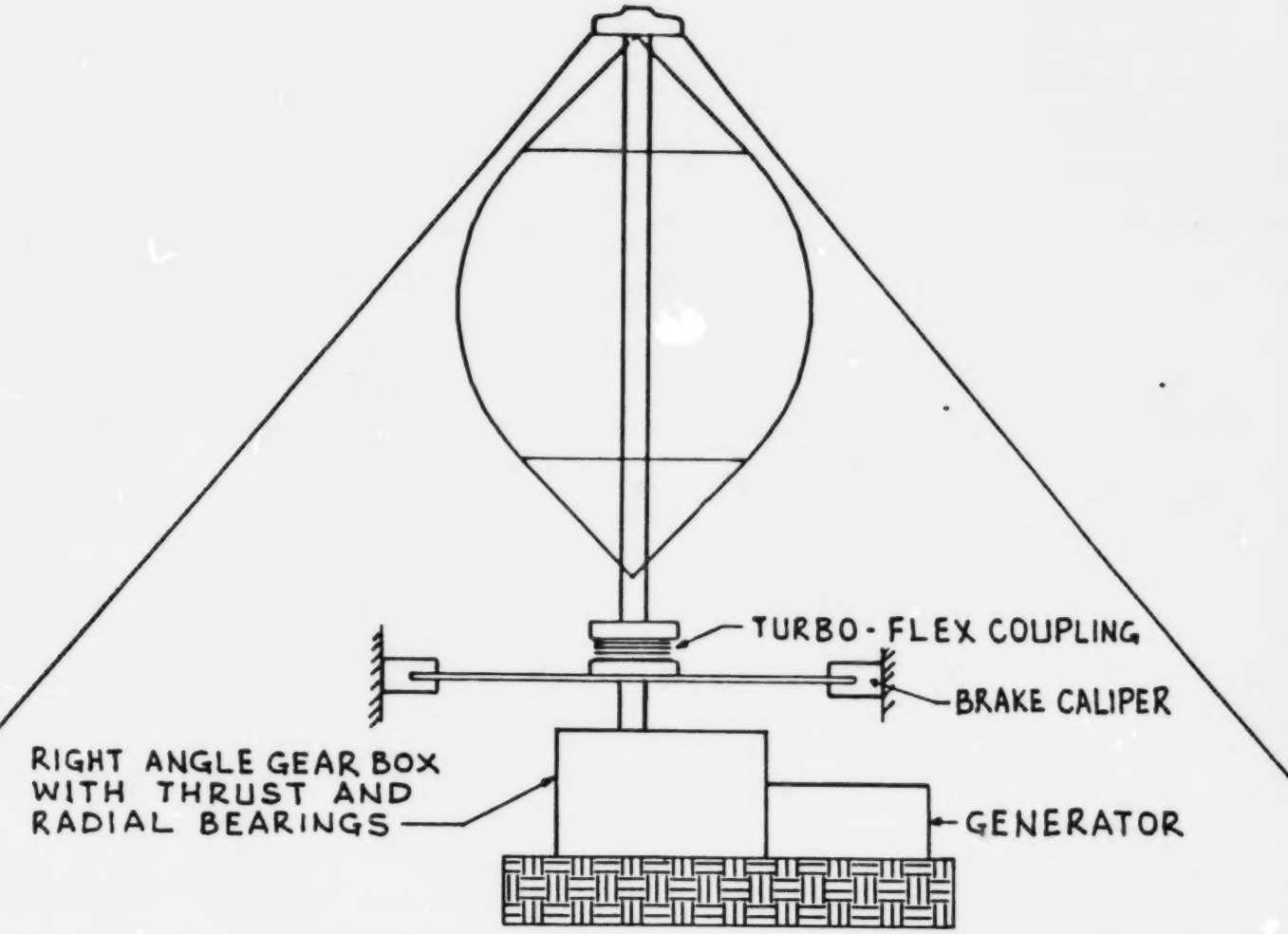



Figure 3.5(c) Alternative Drive Train Configurations.

Configuration (c)

The final arrangement considered is shown in Figure 3.5(c). Its main feature is that the lower bearing is built into the gearbox, allowing a more compact design and the elimination of intermediate couplings and obviating the need for a separate support to the bearing.

Disadvantages of this system are the specialized nature and design of the gearbox-bearing and the need to allow temporary support to the rotor in the event that the gearbox is replaced. In addition some support structure must still be provided for the brakes.

Braking Systems

Three different braking systems were considered: mechanical brakes, aerodynamic brakes and electrical braking. The selection may depend on code requirements for a particular locality but this is speculative and was not considered in this study.

Aerodynamic brakes would be similar to those presently used on Eole and consist of flaps which are deployed at the mid-rotor section of the blades. It has been found from experience with Eole, and also with earlier Indal machines, that such brakes are not reliable, are difficult to maintain, and add considerable mass, inertia and cost. They have not been proven to be cost effective and were rejected in this Study.

Electrical braking is possible only with a variable speed power conversion system and must be accompanied by a set of resistors for dumping of power when the utility grid connection fails. However, if such an electrical system is included then braking can be carried out in a controlled and benign manner and with minimal loads.

Mechanical brakes must be included if only as a back-up to the electrical brakes, for the final slowing of the rotor and as a parking brake. If electrical braking is not included then the mechanical brakes become the primary braking system. These brakes can be located on the low speed shaft above the main bearing (as shown in Figure 3.5), or below the bearing (possible in Figure 3.5(b) only) or they may be located on the high speed shaft between the gearbox and the generator. The last of these locations implies that the braking torque must pass through the gearbox adding to the loads on the gearbox and reducing the reliability of the system. The reduced cost of high speed brakes must, therefore, be balanced against an increased service factor and final cost of the gearbox.

Most mechanic brake systems have employed a single disc with several spring activated, hydraulically restrained calipers. Modifications of this combination may include a pneumatic instead of a hydraulic system and floating calipers to track movement of the disc. At least two calipers are required for load balancing and four calipers allows a second pair to act as an independent and/or emergency system. If a hydraulic system is used, care must be taken in the design to ensure that no hydraulic fluid contaminates the brake disc.

For Milestone Two report the mechanical brakes were assumed to be located on the low speed shaft above the lower bearing. In Section 5 the option of locating them on the high speed shaft is discussed.

## 4.0 PRELIMINARY DESIGN

This section presents some of the findings from the preliminary analysis and design of the two rotors which are reported in full in Milestone Three report.

### 4.1 Design Criteria

The design criteria for the rotors were presented in Milestone Two report and are based on those defined in CSA F416-87 (Wind Energy Corrosion Systems - & Safety, Design and Operation Criteria). The basic loading cases are:

a) dead load of all components;
b) survival wind on all exposed surfaces. For this study a peak wind speed of 65 m/s (at mid rotor) was adopted;
c) ice coating on all exposed surfaces of 50 mm (with a density of 900 $kg/m^3$);
d) static and cyclic loads transmitted from the rotor during normal operation;
e) loads associated with a 10% overspeed;
f) maximum static and cyclic loading in a 3 second guest while in operation;
g) loads experienced during normal starts and stops;
h) loads experienced during emergency stops;
i) loads experienced during transportation and installation;
j) seismic loading

The combination of the loads considered are -

1) a+d (rated power);
2) a+d+g (normal starts and stops at rated power);
3) a+b (survival wind);
4) a+0.5b+c (full ice and partial wind acting on iced members);
5) a+e+f (overspeed and gust);
6) a+e+h (overspeed and emergency stop);
7) a+e+f+h (overspeed and gust and emergency stop).

Seismic and transportation loads have been neglected in this study. A "flat terrain" surrounding was assumed so that a wind shear exponent of 0.16 was used (0.07 for survival wind), and a roughness height of 0.001m. A cutoff windspeed of 18 m/s at mid rotor has been assumed and a gust speed of 22.5 m/s.

The load factors adopted in the design process are -

| | |
|---|---|
| . dead loads and inertial loads | 1.25 |
| . survival wind loads | 1.25 |
| . other aerodynamic loads | 1.50 |
| . fatigue loads | 1.00 |

A resistance factor of 0.9 for members and 0.67 for connections is normally adopted. Fatigue strengths reflect the mean strength less two standard deviations in the operating environment.

Design for Fatigue

Fatigue strength is a governing criterion for many of the components and details of the rotor. An accurate design require detailed information on both the loading spectrum (over 20 years) and the fatigue strength in the environmental conditions to be expected.

The application of this procedure to all rotor components is time consuming and a shortened procedure was adopted in order to evaluate preliminary designs. The shortened procedure requires that the rms, or standard deviation, of the cyclic stresses be estimated for a mid-rotor windspeed of 18 m/s (the selected cutout speed) and that stress cycles with a full range of 4.0 times the rms should be less than the constant amplitude fatigue strength at $10^7$ cycles. This procedure recognizes that random variations in the stresses cause some excursions of up to about 6.0 times the rms (Reference 2) but also that these cycles are less common and that the rotor does not run constantly in high winds.

## 4.2 Blade Design

### Rotor #1

This rotor has a single chord and airfoil which is similar to the design used for Eole. It is illustrated in Figure 4.1 and is made up of a reinforced steel box with leading and trailing edge fibreglass fairings. The blade is made up of a number of straight lengths with kinks at the splice connections There are ten such splices.

The properties of the steel blade used in the finite element analyses are listed in Table 4.1 together with information on the steel properties. The masses of the splices have been estimated and included in the uniform mass of the blade.

### Rotor #2

This rotor has blades which are constructed from fibreglass pultrusions which are illustrated in Figures 4.2(a) and (b). Each section is manufactured from several straight pultrusions which are bonded together to form a complete airfoil.

The straight blade lengths formed in this way will be spliced together using bonded fibreglass doubler plates as illustrated in Figure 4.3. The complete blade length will then be elastically bent into place and attached to the two struts and the two root plates. The blade profile, so formed, will not be

exactly the same as the corresponding troposkein but the troposkeins are rarely used in practice and gravity forces always introduce some flatwise blade bending.

In order for the process of elastic bending to be possible without overstressing the fibreglass and vinyl ester composite it is necessary that the material be sufficiently flexible and be able to withstand the required strain. The maximum strain estimated for the blade of this rotor is approximately 0.005. This strain would cause yielding in materials such as steel and aluminum, but is about one third of the strain that will cause failure in a low-density fibreglass pultrusion. The sections shown in Figure 4.2 have been configured to avoid the possibility of local buckling due to the compression involved in the elastic bending process.

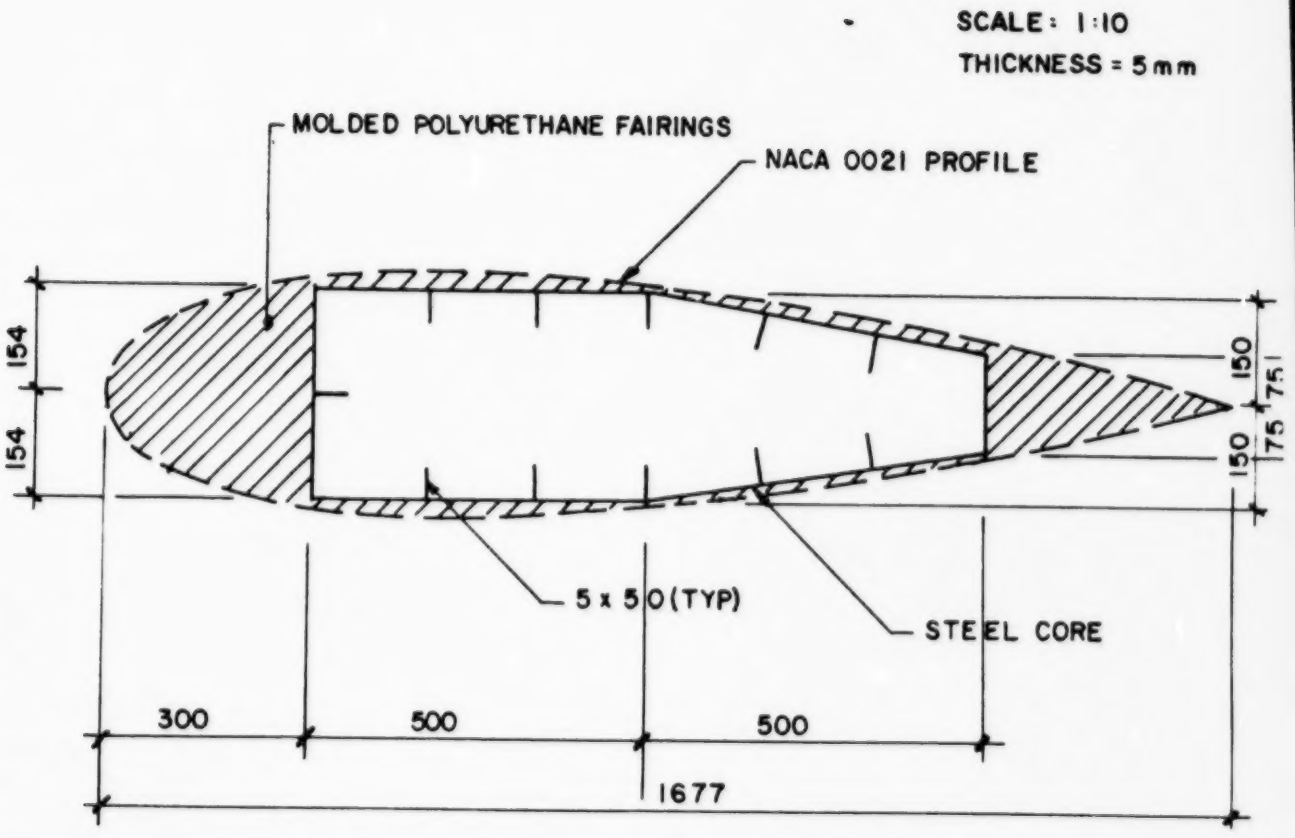



Figure 4.1 Rotor #1. Blade Section.

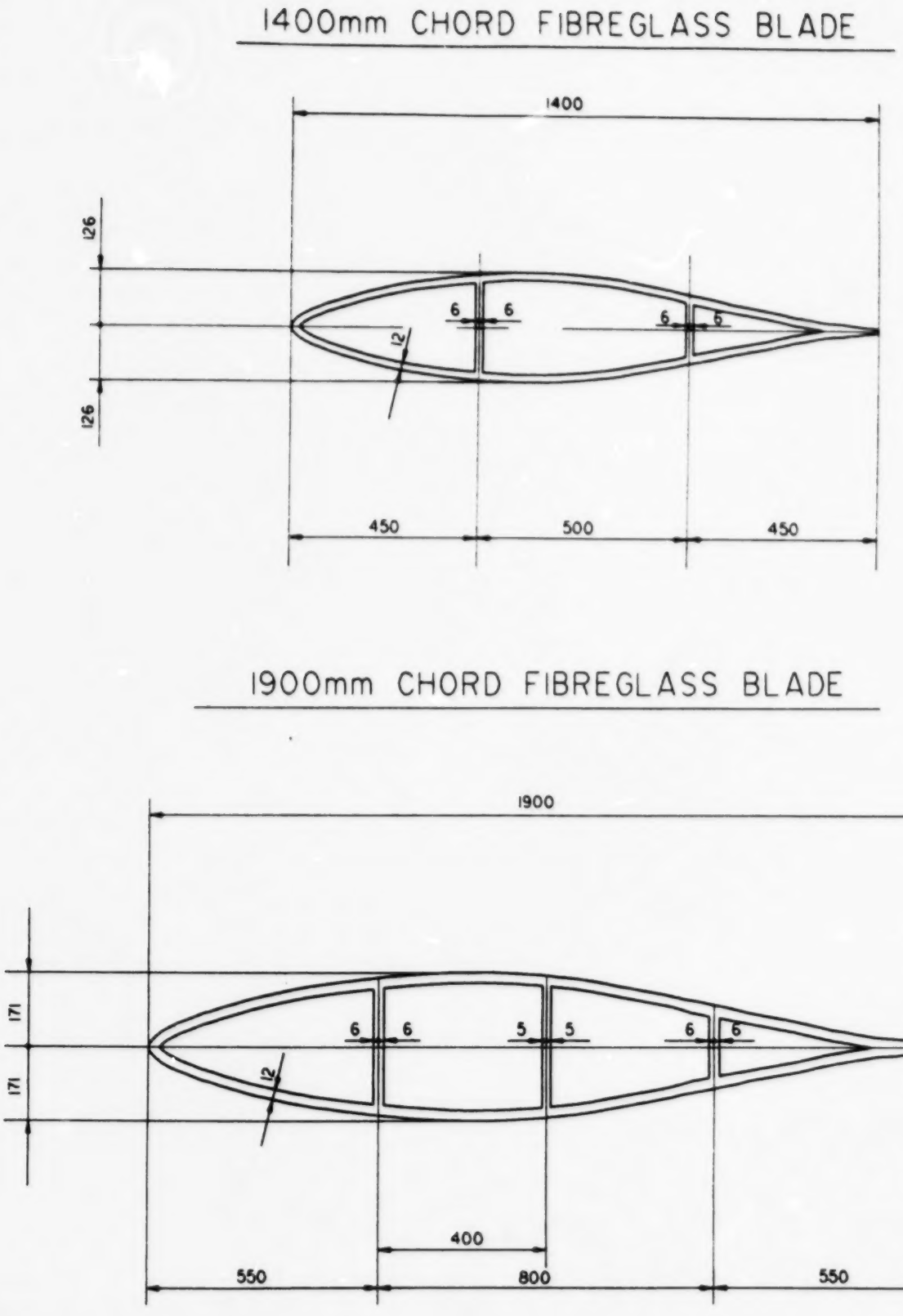



**Figure 4.2 Rotor #2 - Blade Sections**

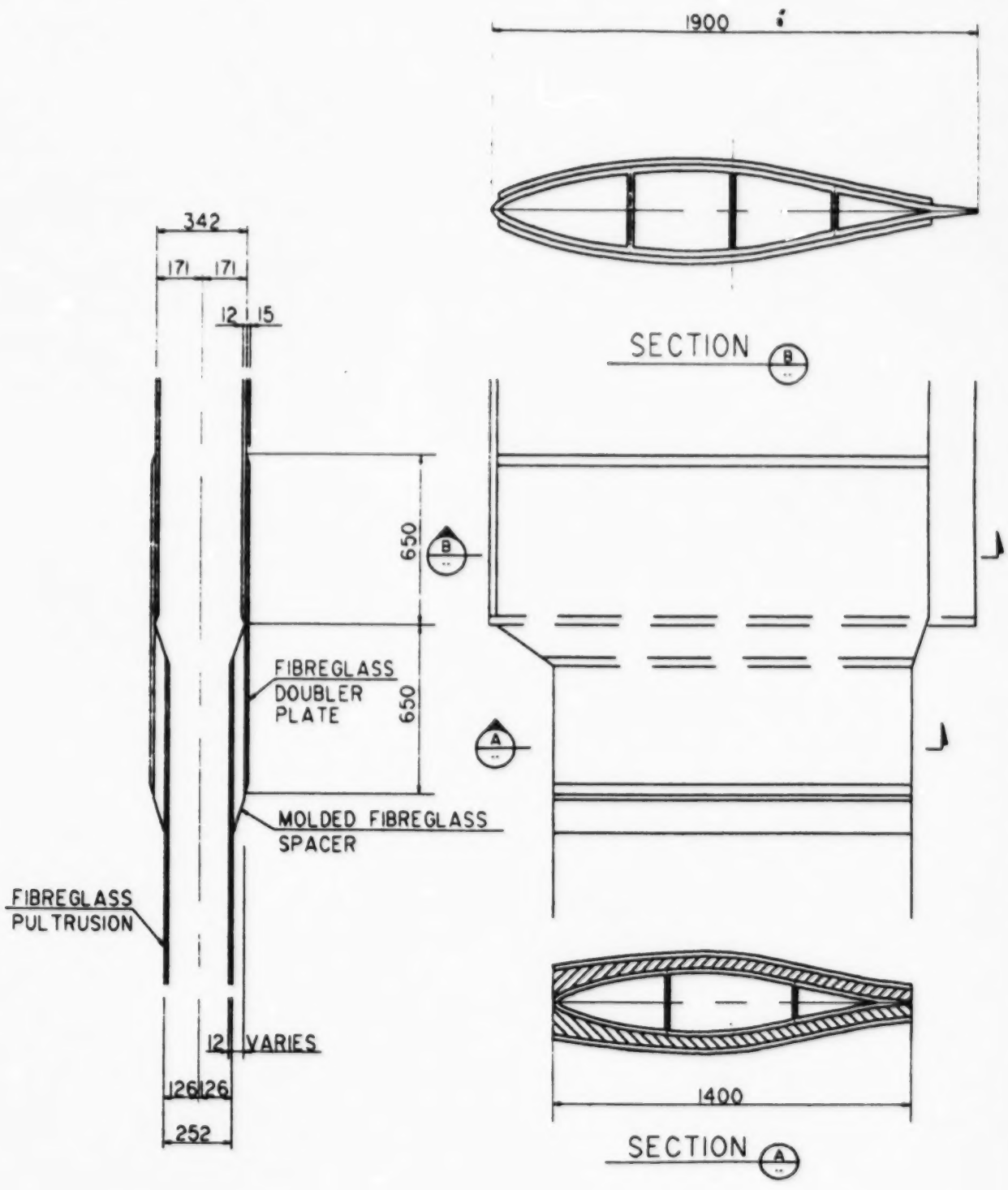



Figure 4.3 Rotor #2 - Bonded Splice Connection

TABLE 4.1

GEOMETRIC AND MATERIAL PROPERTIES OF BLADE SECTIONS

| | ROTOR #1 | ROTOR #2 | ROTOR #2 |
|---|---|---|---|
| Total Chord, mm | 1667 | 1400 | 1900 |
| Structural Section Area, $mm^2$ | 15000 | 37900 | 51400 |
| Moment of Inertia $I_{xx}$, $mm^4$, X $10^6$ | 196 | 300 | 550 |
| Moment of Inertia, $I_{yy}$, $mm^4$ X $10^6$ | 1400 | 7760 | 14300 |
| Polar Moment of Inertia, J, $mm^4$ X $10^6$ | 567 | 722 | 1330 |
| Mass Density, $kg/m^3$ | 7850 | 1800 | 1800 |
| Structural Mass, kg/m | 117.5 | 68.2 | 92.5 |
| Non-Structural Mass, kg/m | 36.5 | 0 | 0 |
| Elastic Modulus, MPa | 200000 | 17000 | 17000 |

## 4.3 Column Design

The central column plays a major role in maintaining the structural integrity of the rotor and also in determining the natural frequencies of the rotor. Most Darrieus rotor columns have comprised a circular steel tube which, on smaller rotors, has been of spiral welded construction. Larger rotors and larger tubes, such as that of Eole, have included circumferential stiffening of the walls to ensure local stability.

The diameter of the column is dictated largely by stiffness requirements and the need to maintain the fundamental natural frequency above the 1 per revolution excitation frequency. This led to a column diameter of 3.500 m and a wall thickness of 8 mm for the first (conventional) rotor.

The second "lightweight" rotor design is not so constrained since the fundamental natural frequency is designed to be less than the 1P excitation. The diameter of the tube for this rotor was, therefore, selected at 2.50 m diameter with a wall thickness of 7.0 mm. This reduced column diameter together with the decreased axial thrust from the guy cables should allow the stiffening requirements to be reduced or eliminated.

## 4.4 Strut Design

The horizontal struts are important in supporting the blades, controlling the natural frequencies, and transferring some of the aerodynamic torque into the central column. The struts of rotor #1 have been designed to be similar to those incorporated in Eole, and are shown in Figure 4.4. They are, essentially, steel trusses and are connected rigidly to the steel-core blades.

The struts for rotor #2 are illustrated in Figure 4.5. They are also steel trusses but of a much lighter construction and are connected to the blade through a horizontally-orientated pin.

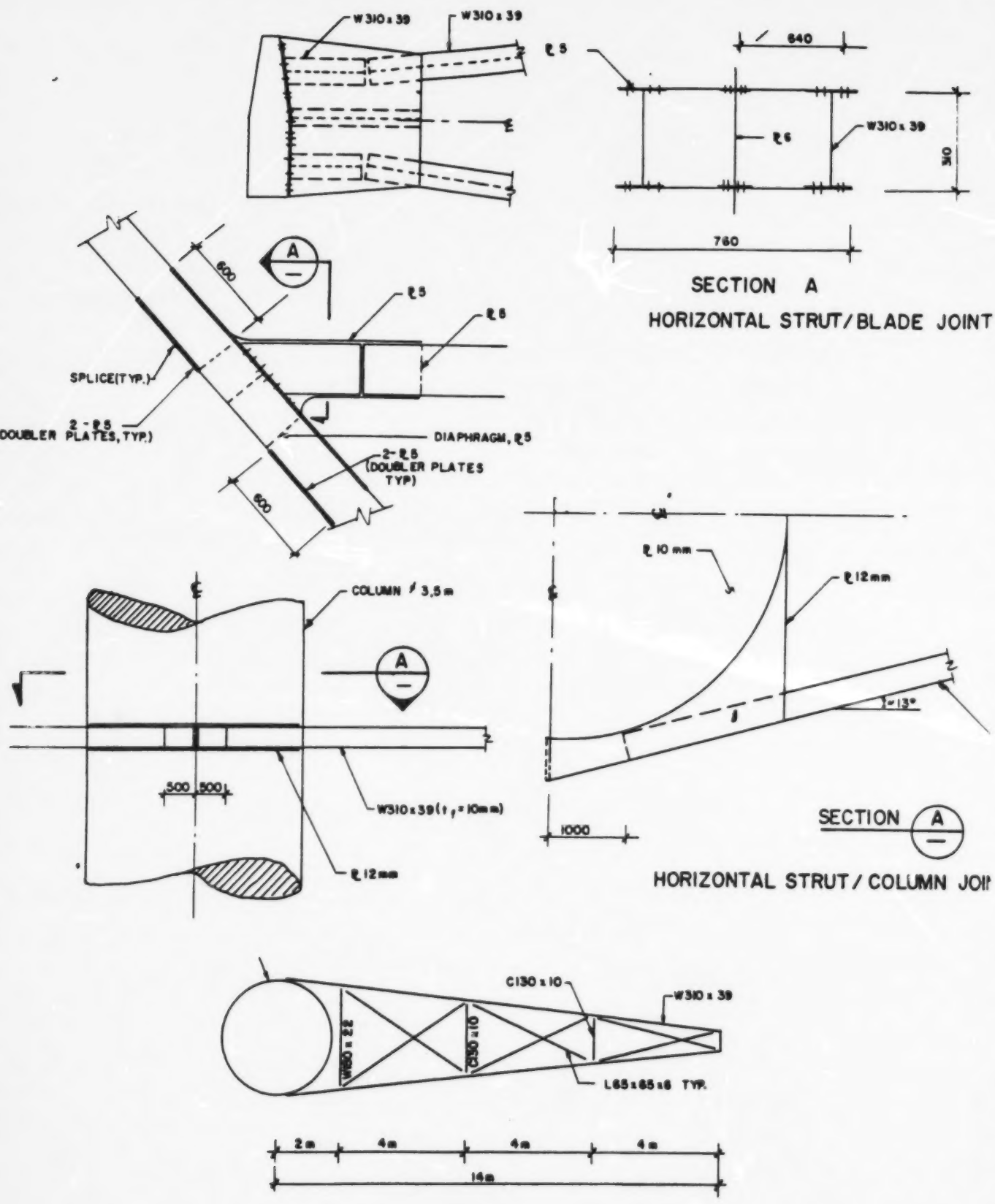



Figure 4.4 Rotor #1 - Horizontal Struts and Connections

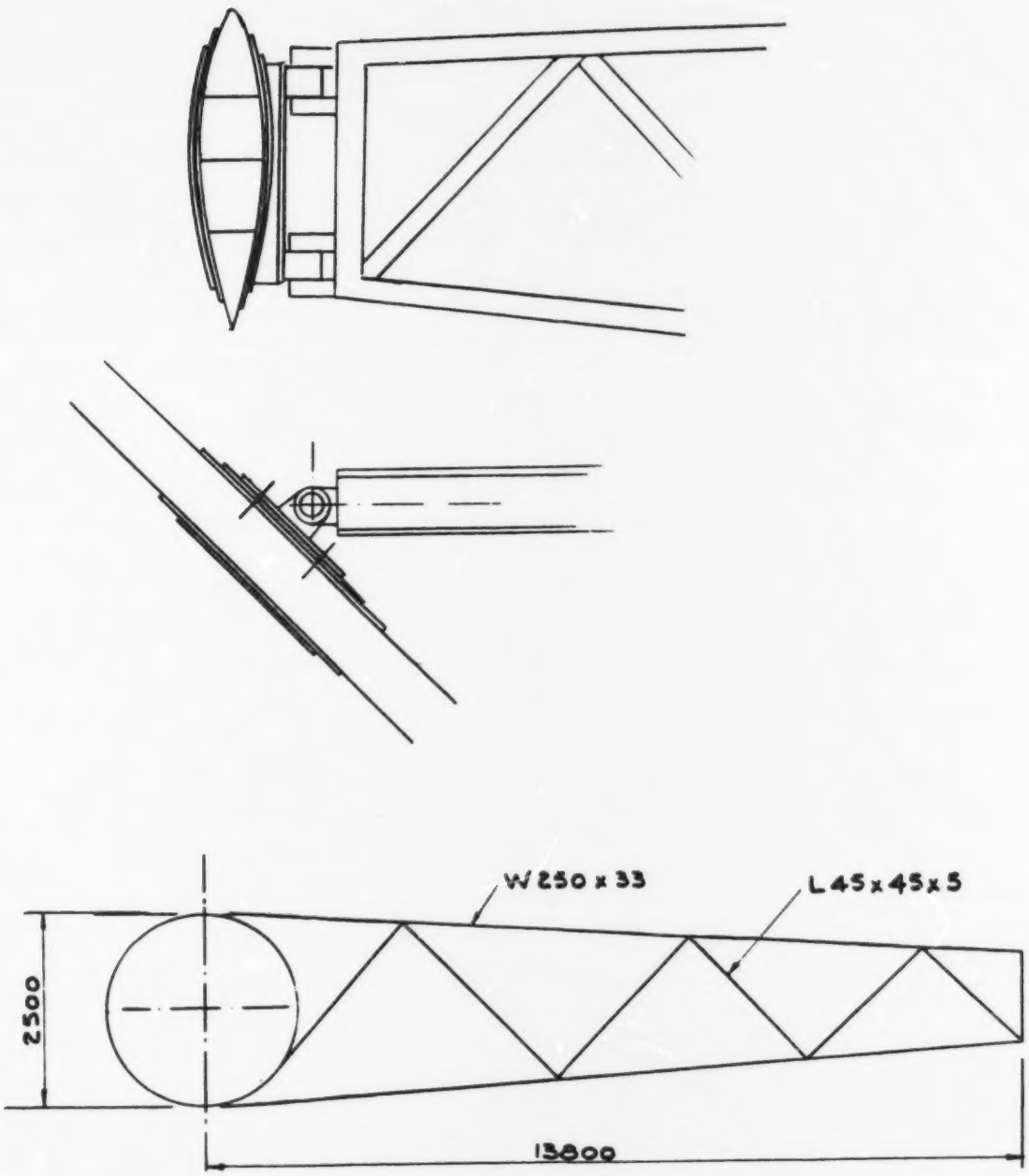



Figure 4.5 Rotor #2. Horizontal Struts and Connections

## 4.5 Natural Frequencies

Natural frequencies of the two rotors were calculated using NASTRAN and also the specialized program Eole (4, 5) which is designed to run on an IBM PC-AT. The results of calculations at a number of rotor speeds are summarized on the fan plots shown in Figures 4.6 and 4.7. These diagrams indicate possible resonant conditions where natural frequencies coincide with excitation frequencies at multiples of the rotor speed.

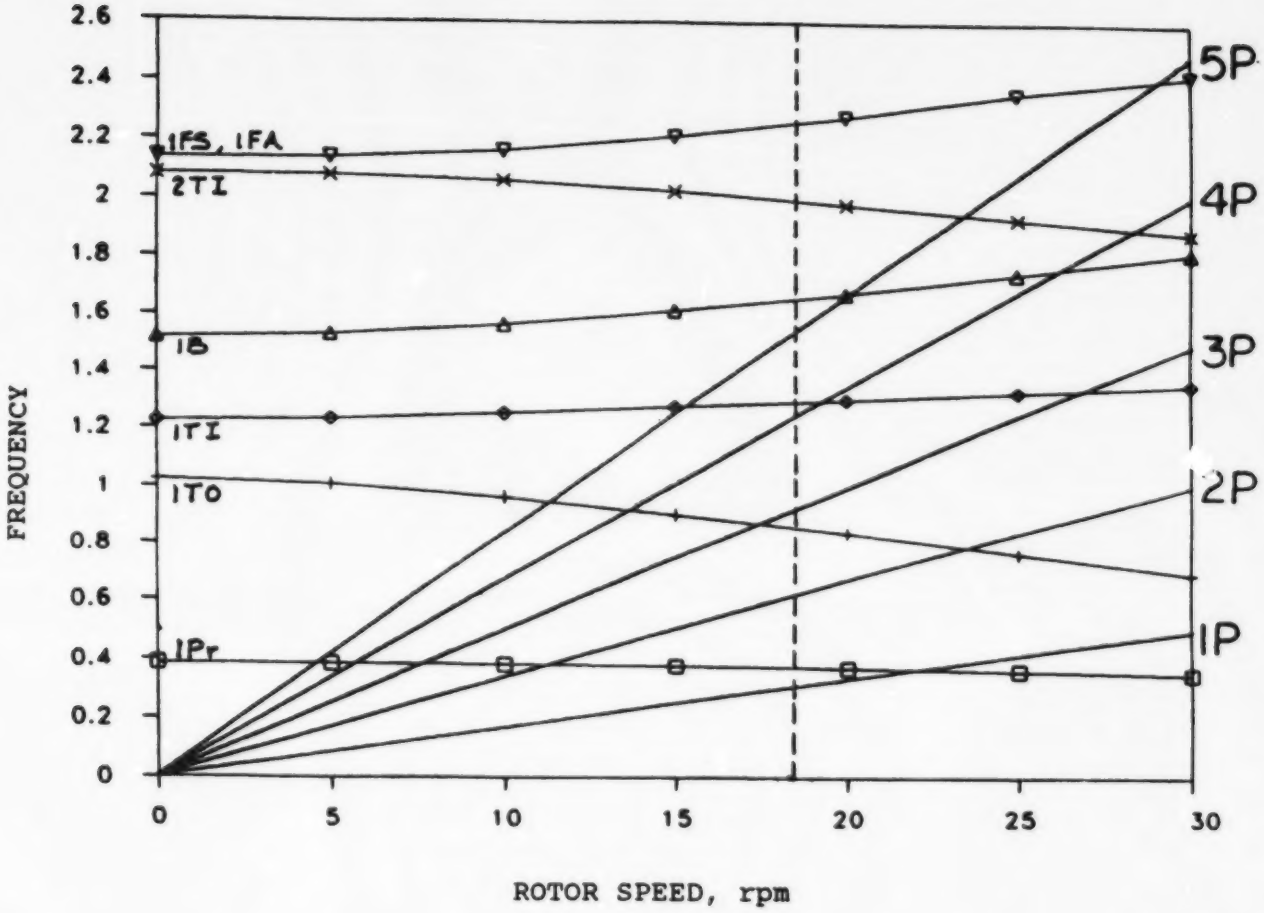
The fan plots for the two rotors are very different, reflecting the different philosophies behind each design. In general the natural frequencies of the lightweight rotor are lower than the corresponding values of rotor #1. This is especially true for the fundamental natural modes (1TO and 1TI) which, for the lightweight rotor, are governed by the stiffness of the guy cables. The two straight lines, 1TO and 1TI, emanating from one point are a consequence of the Coriolis action associated with the rotation.

Coriolis action couples in-plane motion with out-of-plane motion (and vice versa) so that none of the mode shapes at 0 rpm remain pure. However it is possible to predict which resonance crossings are likely to be dangerous. Those modes which involve propeller or symmetric in-plane action will be excited by even harmonics (2P, 4P) whereas all other will be excited by the odd harmonics (1P, 3P, etc.).

The steel-bladed rotor #1, operating at 18.5 rpm, is close to a 3P/1TO resonance (at about 17 rpm) and also a 5P/1B crossing at 20 rpm. The latter might become serious in the event of an overspeed but excitation of 5P is not great. It is apparent that response of this rotor might be improved if all natural frequencies were lowered by making the rotor less stiff.

Figure 4.7 shows that rotor #2 is also free of serious resonance problems at 23.5 rpm but there is a 3P/1B crossing at about 27 rpm which might become active in an overspeed. The fundamental 1TO mode is very low (0.08 Hz at 23.5 rpm, well below the major 1P excitation at 0.392 Hz so that a dynamic amplification less than unity might be expected. However, the dynamics of a one degree of freedom gyroscopic system are different from conventional systems. It can be shown that in the absence of damping the dynamic amplification factor (DAF) of a cyclic load at 1P (in the rotating frame) in the direction of the load is given by

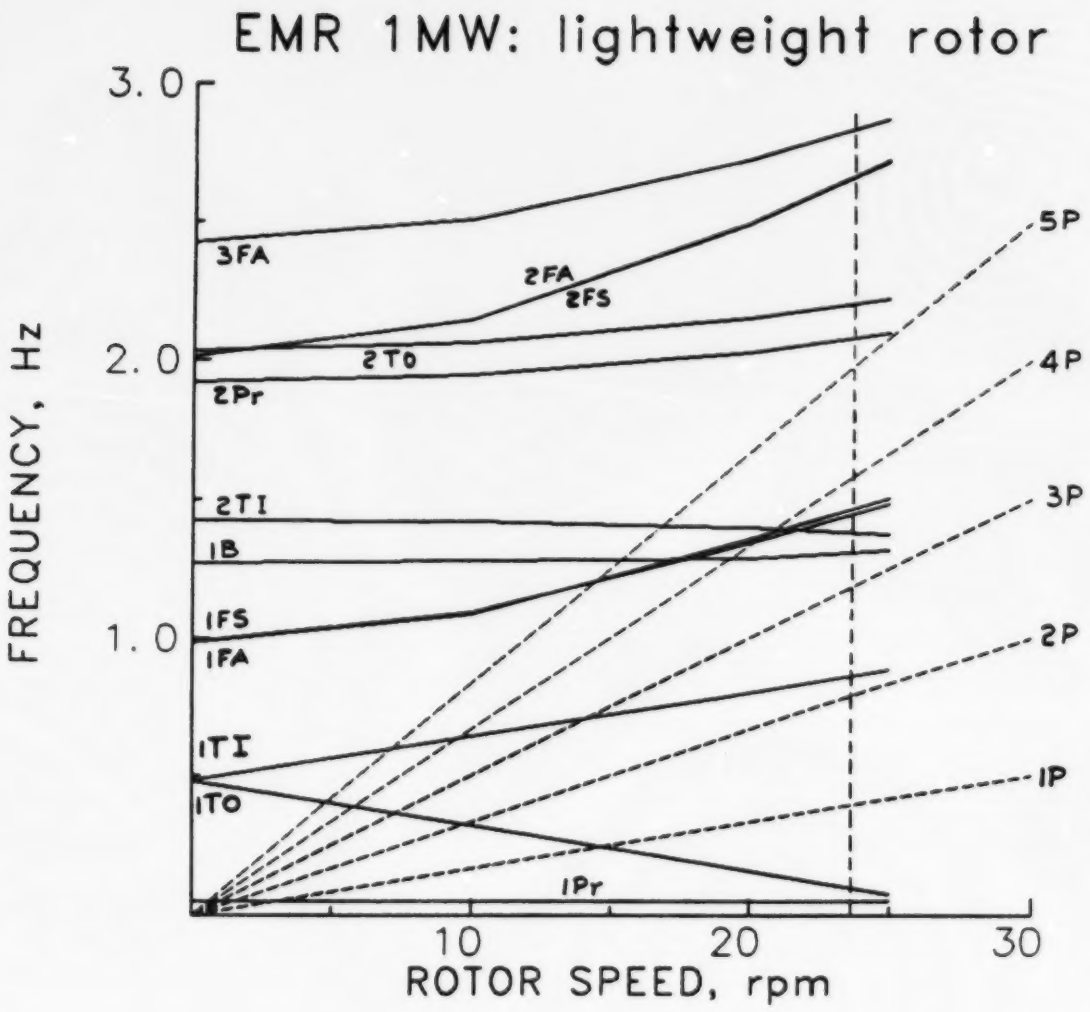
$$DAF = \frac{(2\beta^2 - 1)}{(4\beta^2 - 1)}$$




Key:

Pr.....propeller
T.....tower
B.....butterfly
F.....blade flatwise
I.....in-plane
O.....out-of-plane
S.....symmetric
A.....asymmetric

Figure 4.6 Rotor #1. Natural Frequencies vs. Rotor Speed



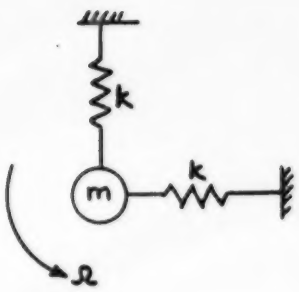

Key:

Pr.....propeller
T.....tower
B.....butterfly
F.....blade flatwise
I.....in-plane
O.....out-of-plane
S.....symmetric
A.....asymmetric

Figure 4.7 Rotor #2. Natural Frequencies vs. Rotor Speed

response of rotating system to 1P excitation

natural frequency $= \sqrt{\frac{k}{m}} = \omega$
rotational frequency $= \Omega$
excitation frequency $= \Omega$



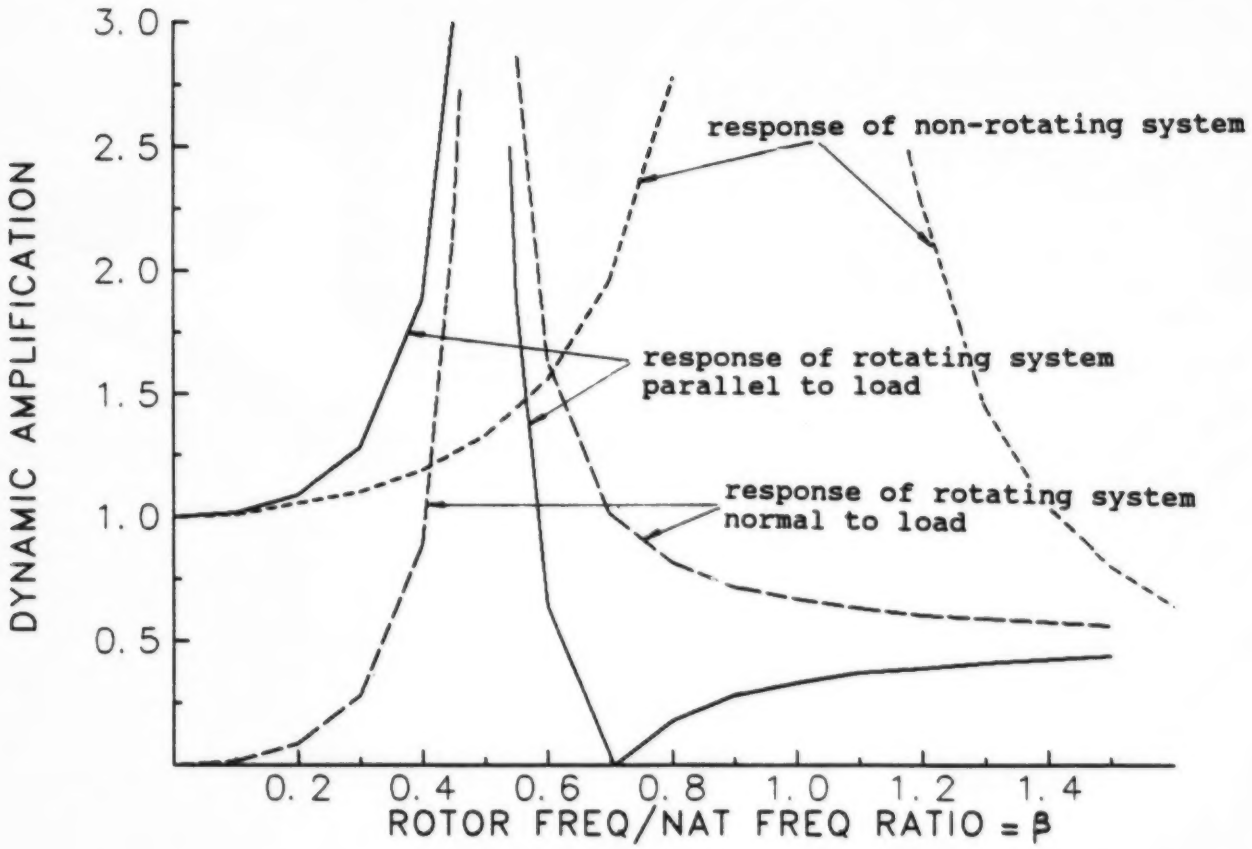




Figure 4.8 Dynamic Amplification of a Rotating System.

Where $\beta = \Omega/\omega$ and $\omega$ = natural frequency of the non-rotating system. In the direction normal to the applied load the dynamic amplification factor is -

$$DAF = \frac{2\beta^2}{(4\beta^2 - 1)}$$

These two curves, together with that for a non-rotating system are shown in Figure 4.8 and demonstrate that beyond the resonance at $\beta = 0.5$, the response is greater in the direction normal to the applied load. This was confirmed by the frequency response analysis carried out on rotor #2.

TABLE 4.2

SUMMARY OF ANALYSIS RESULTS

| | ROTOR #1 | | ROTOR #2 | |
|---|---|---|---|---|
| *Gravity And Ice | | | | |
| axial force in blade at lower strut | -134 | kN | - 33.2 | kN |
| blade bending moment at lower strut | 744 | kN.m | 188 | |
| maximum blade compression | 65.7 | MPa | 49.4 | MPa |
| *Centrifugal Forces | | | | |
| blade axial force at lower strut | 459 | kN | 257 | kN |
| blade bending at lower strut | 81.7 | kN.m | 17.5 | kN.m |
| maximum blade tension | 62.6 | MPa | 17.7 | MPa |
| *Operation at V=20 m/s | | | | |
| max. rms of cyclic blade stress | 20.4 | MPa | 10.6 | MPa |
| max. rms of cyclic column stress | 8.73 | MPa | 40.0 | Mpa |

## 4.6 Analysis Results

Some analysis was carried out on both rotors for the following conditions

- gravity and ice loading
- centrifugal forces
- operating forces at V = 20 m/s
- 65 m/s survival wind.

Full details of these analysis and their results are given in the Milestone Three report. Some results are summarized in Table 4.2 and additional comments are given below.

1) Rotor #1 is approximately twice as heavy as rotor #2 and all inertial and gravity forces are greater.

2) The dynamic response of rotor #1, as represented by its fanplot (see Figure 4.6) is conventional and is a fairly stiff structure. It may be possible to reduce the size and preload in the guy cables without an appreciable change in the dynamic behaviour.

3) The dynamic response of rotor #2 (see Figure 4.7) is different because it is lighter and softer especially in the fundamental tower modes. This has successfully reduced the 1P response in all components, including the guy cables.

4) The high 3P response predicted for rotor #2 in a 20 m/s windspeed is, judging by field measurements, probably conservative. Furthermore this dynamic response can be reduced by tuning of the rotor structure.

5) The effect of turbulence on the operating rotors has not been included. From previous experience the effect of turbulence can be to increase the rms of the flatwise blade bending. Details design and analysis should include this effect.

6) Rotor #1, being stiff, is very stable in survival winds but the blades of rotor #2 may be prone to buckling under this condition. In addition the soft guy system of rotor #2 will lead to motion of the parked rotor and this should be studied in more detail.

7) The configuration of rotor #2 may be regarded as more efficient since the same swept area is achieved with a shorter column. The change in the slope of the blades at their roots associated with the lower H/D ratio might require careful design and positioning of the upper bearing to ensure adequate clearance between blade and cable. However, the cable slope of $25^{o}$ (in place of $35^{o}$) for rotor #2 avoids this problem as well as achieving the required guy cable properties.

8) More is known about the fatigue strength of high strength bolted connections than about bonded fibreglass pultrusions. There is, therefore, more confidence in the fatigue design of the rotor #1 blades than in rotor #2 blades. Reasonable assumptions have been made based on limited data. Some connection samples should be tested as part of any further pursuit of rotor #2 design.

9) Rotor #2 design is based on the elastic bending of a low-modulus fibreglass pultrusion. This should be confirmed by sample testing and long term creep tests.

10) The combination of low stiffness but high strength required of the guy cables of rotor #2 may be better satisfied by using Kevlar in place of steel bridge strand.

11) Much of the 1P bending stresses in the column of rotor 2 is due to inertial effects associated with the considerable upper bearing displacement. A careful examination of the distribution of these stresses may lead to a more efficient column design for the rotor.

12) The very low fundamental operating natural frequency of rotor #2 should not be a cause for alarm. These frequencies are measured in the rotating frame and could be excited only be turbulence and structural out-of-balance forces.

13) The steel struts of rotor #2 are much stiffer (axially) than the fibreglass blades and this leads to some undesirable effects. Consideration should be given to making the struts from a material of elastic modulus similar to that of the pultrusions.

# 5.0 COST ESTIMATE

## 5.1 Costing Basis

Costs were obtained for all of the components and sub assemblies that would be required for both of the rotors. The following assumptions were made in obtaining and presenting these results.

1) Costs are in March 1990 Canadian dollars; it is assumed that $1 CDN = $0.84 U.S.

2) Costs do not include Federal, Provincial, or Customs Taxes, All of these vary with the source and the conditions of use.

3) Costs refer to the supply of one machine only. Comments concerning possible bulk purchases and manufacture are made later.

4) Costs include the costs of tooling except where otherwise noted.

5) All costs are accurate to within ± 25%.

6) Costs of land and interconnection have not been included.

7) Costs do not include engineering and other professional costs.

All costs are summarized in Table 5.1 and some explanatory remarks are made in the following sub-sections. Costs were obtained using a number of sources: quotations from suppliers, extrapolation from experience with similar construction and estimate of cost per unit mass.

## 5.2 Rotor

The blades of rotor #1 are similar to those of Eole and estimates of fabrication and material costs could be made with some accuracy. The cost of blades for rotor #2 were based on manufacturers' quotations and estimates of the labour involved in assembly in the shop and in the field.

The columns of both rotors are of similar construction - welded steel tubes with the required amount of stiffening. The cost per unit weight is therefore the same for each but since the weight of the column of rotor #2 is approximately one half of that of rotor #1, the total cost is also approximately one half. The final cost of the steel is close to $4/kg which reflects the considerable labour involved in the upper and lower weldments and in the tube wall stiffening.

The struts of both rotors are of standard steel construction although care must be taken to design connections for fatigue loading. It is possible that the struts of rotor #2 can be made lighter and even made from fibreglass pultusions It is hoped that their costs can be reduced.

## 5.3 Drive Train

In Milestone Two report a number of possible drive train configurations were discussed. The costs shown in Table 5.1 are based upon the configuration shown in Figure 5.1 which is used as a basis for comparison with some alternatives. Some of these alternatives are listed in Table 5.2.

The following configurations were considered.

1) As shown in Figure 5.1: a separate bearing and support structure, low speed brakes and a squirrel cage induction generator.

2) As in Figure 5.1 but the brakes are located on the high speed shaft between the gearbox and generator.

3) As in Figure 5.1 but the bearing is build into the right angle gearbox. This is estimated to increase the gearbox cost by 20%. Some structure for temporary support of the rotor is still required.

4) As in Figure 5.1 with a synchronous generator and a frequency converter system.

The selection of a suitable gearbox rating is important but also relies on some professional judgement since this is not a standard application. The gearbox must be selected for both the repeated operational loads and also for the peak loads associated with transient conditions such as starting and stopping. The location of the mechanical brakes influences the extent of transient loads in the gearbox since brakes on the high speed shaft must pass the braking torque through the gearbox. This should result in an increase in the service factor applied in the gearbox selected. In this study it resulted in a service factor increase from 1.5 to 1.9.

Table 5.2 shows that the drive train cost is dominated by the cost of the gearbox which is very sensitive to the value of the low speed torque. This results in a cost reduction of approximately $200,000 for rotor #2. The cost is, likewise, sensitive to the service factor so that the savings of using high speed brakes is totally offset by the increase in gearbox cost.

The combining of the lower bearings with the gearbox (see configuration #3 in Table 5.2) is also a change which appears to have an overall neutral effect on the total cost. Although this arrangement reduces the number of separate components there are disadvantages to this scheme. It reduces the height and energy capture of the rotor unless a column extension is added, it increases torque ripple by avoiding the torsionally soft shaft between a separate bearing and gearbox, and increases the replacement cost of an already-expensive gearbox.

## 5.4 Guy Cables

There is a large difference between the cost of the guy cable systems for rotors #1 and 2. This is due to the soft support system of rotor #2 resulting in cables that are 1/3 of the diameter, or 1/9 of the weight. There are corresponding, but not as dramatic savings to be obtained in the bearings, the housings, the upper head, and supporting weldments.

The cables of both systems are assumed to use steel structural bridge strand construction which offers high strength and stiffness. However, for the design of rotor #2 a cable of low stiffness (but maintaining high strength) is preferred and this may result in alternative cable construction or materials being chosen.

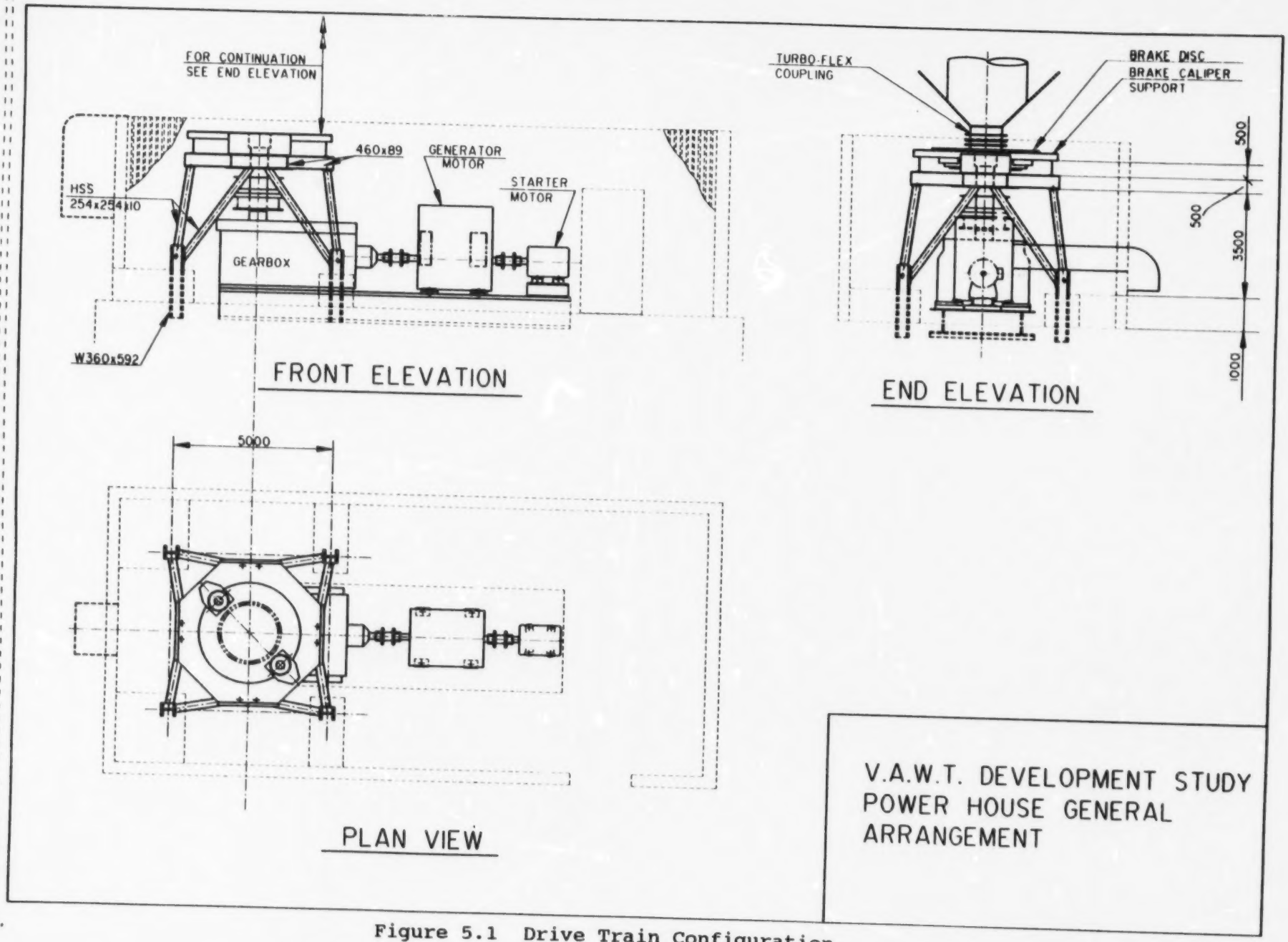



Figure 5.1 Drive Train Configuration.

## Table 5.1

## Summary of Estimated Costs.

| Item | Rotor #1 | Rotor #2 |
|---|---|---|
| . column, upper lower weldments, upper bearing support | $ 256,650 | $ 135,850 |
| . upper head | 13,000 | 9,750 |
| . horizontal struts | 19,700 | 18,300 |
| . blades | 277,350 | 120,000 |
| . upper bearing | 20,000 | 3,000 |
| . guy cables & attachments | 29,600 | 11,000 |
| Rotor Subtotal | $ 616,300 | $ 297,900 |
| . turbo flex coupling | 25,000 | 25,000 |
| . brake system | 70,000 | 52,000 |
| . lower bearing | 30,000 | 20,000 |
| . couplings | 50,000 | 40,000 |
| . gearbox | 482,000 | 305,000 |
| . generator | 50,000 | 50,000 |
| . support structure | 17,000 | 11,000 |
| . enclosure | 60,000 | 50,000 |
| . foundations & anchors | 75,000 | 47,000 |
| . control system | 10,000 | 10,000 |
| . switchgear | 20,000 | 20,000 |
| Drive Train Sub-total | $ 889,000 | $ 630,000 |
| . assembly and installation | 132,100 | 96,500 |
| TOTAL | $ 1,637,400 | $1,024,400 |

TABLE 5.2

Costs of Alternative Drive Train Configurations

| ROTOR | CONFIG. #1 STANDARD | | CONFIG #2 HIGH SPEED BRAKE | | CONFIG #3 BEARING IN GEARBOX | | CONFIG. #4 FREQUENCY CONVERTER | |
|---|---|---|---|---|---|---|---|---|
| | ROTOR #1 | #2 | ROTOR #1 | #2 | ROTOR #1 | #2 | ROTOR #1 | #2 |
| Turbo flex coupling | $ 25,000 | $25,000 | $ 25,000 | $ 25,000 | $25,000 | $ 25,000 | $ 25,000 | $25,000 |
| Lower bearing | 30,000 | 20,000 | 30,000 | 20,000 | --- | --- | 20,000 | 20,000 |
| Low speed brake system | 70,000 | 52,000 | --- | --- | 70,000 | 52,000 | 70,000 | 52,000 |
| High speed brake system | --- | --- | 14,300 | 14,300 | --- | --- | --- | --- |
| Low speed couplings | 50,000 | 40,000 | 50,000 | 40,000 | --- | --- | 50,000 | 40,000 |
| Gearbox | 482,000 | 305,000 | 552,000 | 391,000 | 578,000 | 367,000 | 478,000 | 305,000 |
| Generator | 50,000 | 50,000 | 50,000 | 50,000 | 50,000 | 50,000 | | |
| Frequency converters | | | | | | | 385,000 | 385,000 |
| Support structure | 17,000 | 11,000 | 17,000 | 11,000 | 5,000 | 5,000 | 17,000 | 11,000 |
| TOTAL | $724,000 | $503,000 | $738,300 | $551,300 | $728,000 | $499,000 | $1,045,000 | $838,000 |

## 5.5 Foundations

The type and cost of the power module foundation and of the guy cable anchors will depend on the soil conditions. For the costs given in Table 5.1 "average" conditions were assumed.

There are large differences in the mean loads to be carried by the foundations and anchors of rotors #1 and #2, but the differences between the cyclic loads will be less. The difference between the two sets of costs reflects both of these facts.

# 6.0 SUMMARY

## 6.1 Conclusions

- The costs shown in Table 5.1 are, as explained, incomplete. They do not include possible taxes, professional fees, land purchase and preparation, electrical connection, instrumentation, operation and maintenance. The final installed costs will, therefore, be greater for a single machine, but considerable discounts are obtainable for several machines. An overall discount of 20 - 30% may be expected for more than ten machines.

- The relative costs of rotors #1 and #2 are informative. The rotor of #2 is approximately 50% of that of #1 and the drive train is approximately 70% resulting in a total ratio of 63%.

- These results say two things: one is that the cost of the design of #1 is unacceptably high, and the other is that design #2 does have some chance of being cost effective and competitive with other sources of energy.

- Using a target installed cost of $0.40 U.S. per annual kW/h produced and assuming a 8 m/s Rayleigh wind regime, the total installed cost should be within $1,262,000 CDN. In a windfarm environment this cost might be achieved by rotor #2.

- The single item of largest cost is the gearbox (about 30% of the total cost). Considerable savings can, therefore, be obtained by reducing the service factor, decreasing the low speed torques and by volume discounts.

- The use of a frequency converter and the facility of variable speed is a very expensive luxury (see Table 5.2). It is difficult to justify its inclusion in a commercial machine.

- The placement of brakes on the high speed instead of the low speed shaft would save $50,000 and be more convenient to install and maintain. However, it is balanced by the greater service factor that should be applied to the gearbox.

- Placing the lower bearing within the gearbox greatly simplifies the drive train configuration but has little effect on the total cost.

- The use of the tailored airfoils leads to all of the cost reductions associated with the higher rotor speed. This is primarily in the cost of the gearbox (a $180,000 reduction).

- The lightweight design and dynamically soft rotor is responsible for any reductions in the rotor costs (approximately $320,000).

## 6.2 Recommendations

The following recommendations are made.

(1) A 1 MW Darrieus rotor using steel-blade construction and a dynamically stiff rotor is not likely to be cost effective.

(2) The concepts incorporated into design #2 should be proven on a smaller and less expensive scale.

(3) Development should concentrate on the manufacture of suitable pultrusions and the fatigue strength of bonded joints.

(4) A small rotor should be modified to investigate the concept of the soft cable support system.

(5) The most immediate gains can be obtained by increasing the rotor speed for a given swept area by the use of tailored airfoils.

## 7.0 REFERENCES

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2 D.E. Berg, P.C. Klimas, and W.A. Stephenson, "Aerodynamic Design and Initial Performance Measurements for the Sandia 34m Diameter Vertical Axis Wind Turbine", Proceedings 9th ASME Wind Energy Symposium, Ed. D.E. Berg, January 1990, New Orleans.